Illinois U Library

Transactions

of the

ASME

History and Performance of the Oil-Refinery Steam-Electric Generating Stations of the Pacific Gas and Electric Company . . . . *V. F. Estcourt* 209

Steam-Electric Power Expansion in Southern California . . . . *W. L. Chadwick* 223

Postwar Planning for Steam Capacity in Northern California . . . . *C. C. Whelchel and W. R. Johnson* 237

The Design of Sewaren Generating Station and No. 1 Unit at Essex Station, Public Service Electric and Gas Company . . . . *F. P. Fairchild* 247

Operating Characteristics of the 100,000-Kw Essex Turbine Generator . . . . *Stanford Neal and V. S. Renton* 267

Heat-Rate Test Results of the 100,000-Kw Essex Turbine Generator . . . . *V. S. Renton and Stanford Neal* 285

Frictional Characteristics of O-Rings With a Typical Hydraulic Fluid . . . . *L. E. Cheyney, W. J. Mueller, and R. E. Duval* 291

Strength-Variance Studies of Plastics . . . . *W. J. Gailus, Steven Yurenka, and A. G. H. Dietz* 299

Theoretical Considerations on the Optimum Adjustment of Regulators . . . . *P. Hazebroek and B. L. van der Waerden* 309

The Optimum Adjustment of Regulators . . *P. Hazebroek and B. L. van der Waerden* 317

The Exhaust-Heated Gas-Turbine Cycle . . . . *D. L. Mordell* 323

Properties and Characteristics of Fuel Oils for Industrial Gas-Turbine Usage . . . . *D. P. Heath and E. Albat* 331

Combination of Heat Sources and Sinks by the Method of Superposition . . . . *F. H. Bridgers* 341

APRIL, 1950

VOL. 72, NO. 3

Transactions

of The American Society of Mechanical Engineers

Published on the tenth of every month, except March, June, September, and December

OFFICERS OF THE SOCIETY:

James D. Cunningham, *President*
Joseph L. Kopf, *Treasurer*
C. E. Davies, *Secretary*

COMMITTEE ON PUBLICATIONS:

Ronald B. Smith, *Chairman*
John Haydock
George R. Rich
C. B. Campbell
Paul T. Norton, Jr.
H. G. Wenig
D. R. Thomas } *Junior Advisory Members*
George A. Stetson, *Editor*
K. W. Clendinning, *Managing Editor*

REGIONAL ADVISORY BOARD OF THE PUBLICATIONS COMMITTEE

Kerr Atkinson—I
Otto de Lorenzi—II
W. E. Reaser—III
F. C. Smith—IV
Hendley Blackmon—V
R. E. Turner—VI
R. G. Roshong—VII
M. A. Durland—VIII

Published monthly by The American Society of Mechanical Engineers. Publication office at 20th and Northampton Streets, Easton, Pa. The editorial department is located at the headquarters of the Society, 29 West Thirty-Ninth Street, New York 18, N. Y. Cable address, "Dynamic," New York. Price $1.50 a copy, $12.00 a year for Transactions and the *Journal of Applied Mechanics;* to members and affiliates, $1.00 a copy, $6.00 a year. Changes of address must be received at Society headquarters three weeks before they are to be effective on the mailing list. Please send old as well as new address.... By-Law: The Society shall not be responsible for statements or opinions advanced in papers or ... printed in its publications (B13, Par. 4).... Entered as second-class matter March 2, 1928, at the Post Office at Easton, Pa., under the Act of August 24, 1912.... Copyrighted, 1950, by The American Society of Mechanical Engineers. Reprints from this publication may be made on condition that full credit be given the Transactions of the ASME and the author, and that date of publication be stated.

History and Performance of the Oil-Refinery Steam-Electric Generating Stations of the Pacific Gas and Electric Company

By V. F. Estcourt,[1] San Francisco, Calif.

Design and operating characteristics are given for the three 1400-psi 940 F steam-electric generating stations built by the Pacific Gas and Electric Company for furnishing process steam and electricity to the three adjacent oil refineries, in addition to generating firm power for its own transmission system. An analysis is made of a number of special problems peculiar to this sort of undertaking, and both the advantages and limitations are discussed and evaluated, with emphasis upon the factors which must be considered in order to gain the maximum benefits from such a joint venture. It is pointed out that this type of plant can deliver the refinery's variable requirements for process steam without materially affecting the ability to generate firm power at high thermal efficiencies. However, high-sulphur fuels require higher exit-gas temperatures in order to avoid air-preheater corrosion, with resultant lower boiler efficiencies.

JOINT studies were conducted in 1937, in conjunction with the Tidewater-Associated, Shell, and Union Oil Companies for the purpose of working out the design details and contractual arrangements whereby the Pacific Gas and Electric Company would construct steam-electric generating stations respectively adjacent to the oil refineries of these three oil companies for the generation of electric power and process steam for the use of the refineries, plus electric power for the transmission system of the Pacific Gas and Electric Company. The three contracts as finally written were almost identical except for certain minor provisions relating to the fuels to be burned and the quantities of steam and power to be delivered to the refineries.

The first of these plants, located at Avon, Calif., adjacent to the Tidewater-Associated Company's oil refinery, was placed in operation on December 15, 1940; the second plant, located at Martinez adjacent to the Shell Oil Company's refinery, went into service on June 13, 1941; and the Oleum Plant, adjacent to the Union Oil Company's refinery, first went on the line on January 15, 1942. A second identical unit was installed at Oleum and placed in service on May 24, 1943. This unit was solely for the purpose of meeting the load growth on the Pacific Gas and Electric system.

A detailed description of the design of these stations already has been published in a paper by C. E. Steinbeck.[2] In that paper was included a plan of the station layout, a cross section of the boilers and turbines, and a flow diagram showing both the conventional plant cycle and the superimposed process cycle for the generation of steam for the neighboring oil refineries. Mr. Steinbeck's paper was written before any of the plants had been placed in operation and therefore was confined to a description of the basic layout. This present paper may be considered as a continuation of the subject from the point where Mr. Steinbeck's discussion ended.

Fundamental Considerations

There are certain fundamental considerations which must be taken into account before it is possible to write a contract which will be satisfactory to both parties on a long-term basis. The refinery, on the one hand, expects to get a cheaper and more reliable source of steam and electric power. On the other hand, if the undertaking is to be attractive to the utility, it must be possible to design the installation so that this sort of service can be given to the refinery at an over-all cost to the utility which is competitive with the cost of generating electric power in its conventional generating stations. This calls for a very careful and realistic analysis in order to avoid overlooking factors which might later prove to be detrimental to either party, or which might result in making commitments which cannot be met in practice, thereby disturbing the balance of the mutual advantages which rightfully can be anticipated in a carefully worked out agreement.

The plant must also be designed not only to meet the existing steam and electric requirements of the oil refinery but must be capable of reasonable expansion in anticipation of future growth not only of the oil refinery but also of the electric facilities of the utility. This may not be easy to do without sacrificing some of the original advantages unless the problem is given detailed consideration in the initial design.

The basic advantages which may be gained by the parties in entering into such an arrangement are as follows:

(*a*) The combined project, if properly conceived, will result in over-all economies which cannot be realized by either party separately. If the various technical and commercial factors are given realistic consideration, the resultant benefits can and should be shared equally by both parties.

(*b*) The oil refinery is provided with a reliable source of low-cost electrical energy and process steam.

(*c*) The oil refinery is provided with a commercial outlet for its waste fuels, and the burning of such fuels for the generation of electric energy is a contribution in the interests of conservation of our natural resources.

(*d*) In addition to the sale of electricity and steam to the oil refinery, an appreciable block of electrical energy with firm power value can be delivered to the transmission system of the utility.

(*e*) A stable source of fuel supply is made available to the utility not only to meet the refinery requirements, but also for the generation of the energy which is delivered into the utility's transmission system.

---

[1] Engineer of Steam Operation, Pacific Gas and Electric Company. Mem. ASME.

[2] "Power and Steam Plants for Oil-Refinery Service," by C. E. Steinbeck, Trans. ASME, vol. 61, 1939, pp. 733–740.

Contributed by the Power Division and presented at the Semi-Annual Meeting, San Francisco, Calif., June 27–30, 1949, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 49—SA-11.

Principal Features of the Contract

The three contracts were signed during 1937 and 1938, and received the approval of what was then known as the California State Railroad Commission but which now functions under the name of California Public Utilities Commission. The maximum steam and electric quantities involved in each case are given in Table 1.

TABLE 1 STEAM AND ELECTRIC REQUIREMENTS OF REFINERIES

Name of company	Process steam, lb per hr	Electricity, kw
Tidewater-Associated Oil Company....	275000	15000
Shell Oil Company..................	325000	15000
Union Oil Company..................	275000	15000

The Pacific Gas and Electric Company agreed to furnish process steam and electric energy in quantities not to exceed the maximum demands given in Table 1, in exchange for certain quantities of "substitute fuels," plus cash payments based upon a demand charge plus a commodity charge. A formula was established wherein the barrels of substitute fuel to be delivered by the oil companies in part payment for the steam and electricity was obtained by multiplying by a certain factor the total pounds of steam or kilowatthours delivered to the refinery. There are different multiplying factors for steam and electricity, respectively.

The definition of substitute fuel, as applied in the contract is "any liquid or gaseous fuel, including refinery-waste fuel such as acid sludge, tars, waxes, and gases which will not leave an unburnable residue in excess of 2 per cent by weight, and will produce substantially the same output of steam from the boilers as when fired to capacity with fuel oil." In addition to the foregoing, a further provision was made at the Oleum Steam Plant to include pulverized petroleum coke having the following analysis:

Volatile constituents..........	not less than 10 per cent
Fixed carbon................	not more than 90 per cent
Ash.........................	not more than 1 per cent
Sulphur.....................	not more than 2 per cent
Impregnated oil..............	not more than 2 per cent

Provision was also made for the purchase by the Pacific Gas & Electric Company of certain quantities of "excess fuel oil" over and above the amounts delivered to it in payment for steam and electricity. This feature provided a firm source of fuel supply for the generation of electric power for the company's own transmission system. In Fig. 1 is given a diagrammatic representation of the disposition of the various quantities of steam, electricity, fuel, and cash involved in the operation of the contract. Based upon a nominal maximum load of 46,000 kw, more than 30,000 kw is delivered to the company's transmission system, and a little under 12,000 kw is delivered to the oil refinery, plus 275,000 lb per hr of process steam. Of the total payment in terms of fuel and cash for these quantities of steam and electricity, 67 per cent is for steam and 33 per cent is for electric energy. Under the operating conditions shown in the diagram, the total payment fuel represents 73 per cent of the total fuel requirements of the station, and the remaining 27 per cent must be purchased under the excess-fuel-oil clause in the contract.

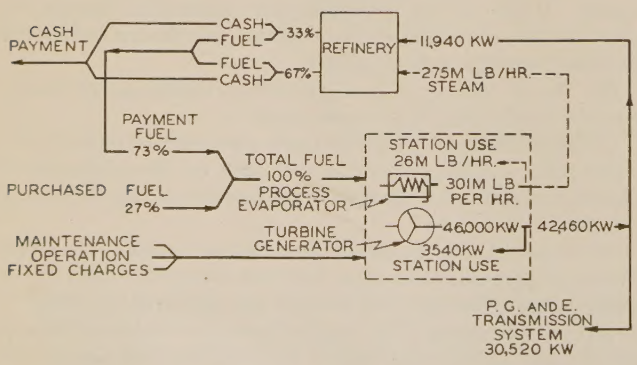


Fig. 1 Distribution of Steam and Electric Loads, Cash and Fuel Payments, etc., Avon Steam Plant

Design and Operating Characteristics

In order to meet the operating requirements of the Pacific Gas and Electric system, the turbine generators had to be capable of providing a firm electrical power source for its own system without being dependent upon the process-steam demands of the oil refinery. It was also necessary to make the installation self-sufficient as an electric generating station even though the contract with the oil refinery were discontinued at some future date. From this point of view, a straight back-pressure turbine presented distinct disadvantages, because in this case there must be an outlet for the exhaust steam before the turbine is able to generate power. Therefore, single 50,000-kw, 3600-rpm tandem-compound units with automatic extraction at the exhaust of the high-pressure turbine were selected for each of the three stations, and are designed for an initial steam pressure of 1400 psig and a total steam temperature of 940 F. Although these units are nominally rated at 50,000 kw, they have under favorable extraction conditions operated as high as 55,000 kw. Their average capability under the usual refinery steam loads is about 46,000 kw.

The turbine is designed so that it may be operated either as a straight condensing tandem-compound machine or as an automatic extraction unit. It is only on rare occasions that it is necessary to operate on a straight condensing basis; and, under these conditions, the capacity of the unit is reduced to approximately 41,000 kw. Thus, should the oil refinery temporarily curtail or discontinue taking process steam, as was the case during the recent Pacific Coast oil-refinery strike, the plant is able to continue generating more than 80 per cent of its nominal capacity. Should the refinery permanently discontinue taking process steam, it would be feasible to add a second low-pressure condensing element of approximately 25,000 kw capacity, thereby raising the nonextraction capacity of the unit from 41,000 to 66,000 kw with the same installed boiler capacity.

The low-pressure turbine exhausts into a condenser which is somewhat smaller than would be used for a conventional condensing machine for the reason that large quantities of steam normally are extracted at the exhaust of the high-pressure element, thus unloading the condenser. This feature is illustrated by the curve in Fig. 2 showing that at 40,000 kw and with a contract demand of 275,000 lb per hr of process steam, the flow to the condenser is only 172,000 lb per hr, as compared with 324,000 lb per hr for straight condensing operation; or stating it in another way, the condenser flow for these conditions, when operating on an automatic extraction basis, is approximately 53 per cent of what it would be for straight condensing operation. Obviously, any reduction in condenser losses with the same kilowatt output results in a substantial gain in economy in the over-all cycle, as well as a saving in initial investment for the smaller condenser. Due to the relatively few hours of operation on a straight condensing basis, it is economical to overrate the condenser for these conditions at some sacrifice in efficiency.

There is a somewhat complicated relationship between the electric generating capability of the turbine generator and the process-steam flow to the oil refinery. From the curves in Fig. 3 it can be seen that, as the refinery steam load increases, the electric load on the turbine generator can be held constant or increased until the control valves on the low-pressure turbine are wide open. When this optimum point is reached, any further increase in the electric load can be made only by increasing the refinery steam load. In this connection it is interesting to note that the second unit installed at Oleum made it possible to divide the extraction load between the two units and thereby operate more nearly at this optimum extraction point, so that the capability of the first machine was slightly increased as a direct result of installing the second unit. At a refinery steam demand of 275,000 lb per hr and with all of the extraction steam coming from one machine, the capacity of the turbine is 46,000 kw, as compared with 48,000 kw on each of two units with the extraction divided equally between them.

The steam which is extracted from the high-pressure turbine is passed through the process evaporator coils, and the condensate is returned to the plant cycle. The heat exchange with the process feedwater results in the generation of steam at approximately 165 psig. The evaporators were selected for a moderate temperature head which varies both with the process steam load and the cleanliness of the evaporator coils. The extraction pressure is controlled by a feeler line connected to the 165-psig steam pipe to the oil refinery. Thus, although the nominal extraction pressure at the exhaust of the high-pressure turbine has been referred to as 240 psig, actually this pressure varies automatically with the evaporator load as indicated in Fig. 4. This serves to hold the temperature head at a minimum, thereby reducing the inherent loss in cycle efficiency where a heat exchanger is interposed, in order to recover as condensate the steam extracted from the high-pressure turbine.

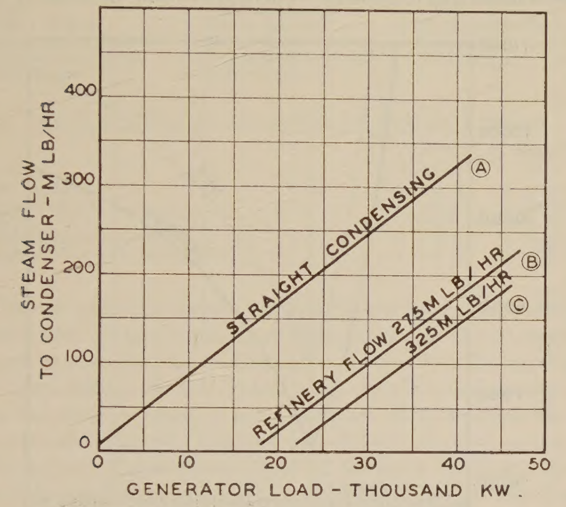


FIG. 2 STEAM FLOW TO CONDENSER FOR DIFFERENT GENERATOR LOADS UNDER BOTH STRAIGHT CONDENSING AND EXTRACTION OPERATION OF 50,000-KW TURBINE-GENERATOR AT AVON STEAM PLANT

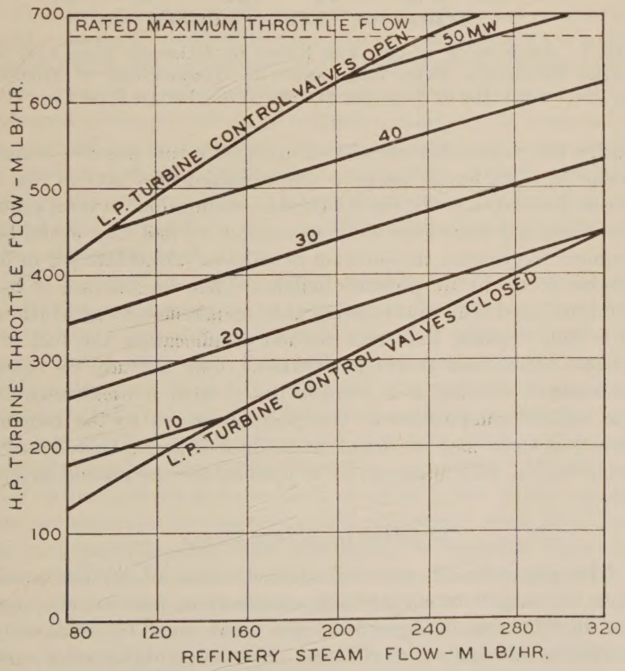


FIG. 3 OPERATING CHARACTERISTICS OF 50,000-KW TURBINE GENERATOR AT AVON STEAM PLANT FOR VARIOUS ELECTRIC AND EXTRACTION LOAD CONDITIONS

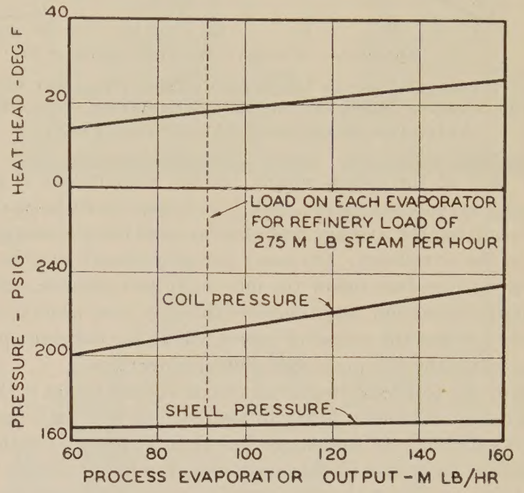


FIG. 4 PROCESS-EVAPORATOR OPERATING CHARACTERISTICS AT DIFFERENT OUTPUTS—AVON STEAM PLANT

In view of the foregoing, the losses resulting from the use of a closed cycle are not as great as sometimes have been indicated. The only other alternative is to deliver the extracted steam from the exhaust of the high-pressure turbine directly to the oil refinery. This immediately introduces a major problem in controlling the chemical content of large quantities of raw make-up water which must be fed to the high-pressure boilers. It is usually difficult to obtain condensate returns from the oil refinery which are free enough from oil contamination for satisfactory high-pressure boiler feedwater. Therefore the added investment cost in water-conditioning equipment plus increased operating and maintenance costs will in some instances offset the theoretical gain in cycle efficiency.

In electric systems such as ours, where an appreciable percentage of the power output is generated from hydroelectric sources, there sometimes occur seasonal conditions wherein all or at least a considerable number of the steam plants must be operated at minimum load in order to prevent loss of water over the dams during periods of high run-off. As indicated in Fig. 5, the amount of process steam required by the refinery determines the minimum electric load which can be carried by the turbine generator when operating on a strictly automatic extraction cycle. This minimum occurs when the low-pressure-turbine control valves are closed so that the high-pressure turbine is functioning as a straight back-pressure machine with its load fixed by the amount of extraction steam being delivered to the evaporator coils. In order to reduce the load below this minimum, it is necessary to by-pass a portion of the steam requirements for the process evaporators around the turbine and through the reducing valves which are provided primarily for occasions when the turbine is shut down for overhaul. Obviously, there is considerable loss in efficiency when

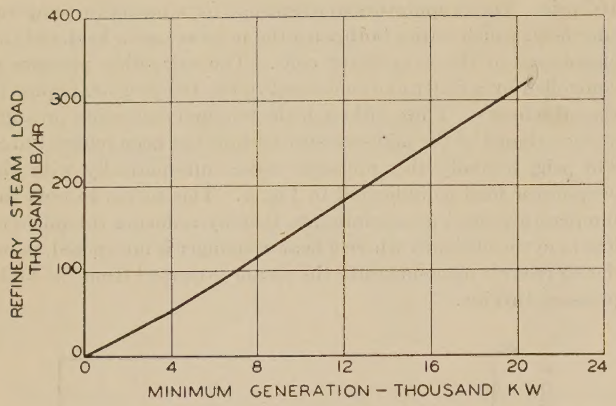


Fig. 5 Minimum Possible Loads on 50,000-Kw Turbine Generator for Various Refinery Steam Loads When Operating on Automatic Extraction—Avon Steam Plant

all or part of the steam supply for the evaporators is passed through the reducing valves, and these factors must be evaluated in determining the average cost of steam and electric energy supplied to the oil refinery. On many power systems it may be feasible never to operate below the minimum load possible with full-automatic extraction, and therefore the only time when it will be necessary to use the reducing valves will be for the short periods during which the turbine is shut down for overhaul.

One of the problems requiring special consideration is the size and number of boilers. Due to the burning of waste fuels the boiler availability is somewhat less than is possible with good commercial fuels, and furthermore, the oil refinery cannot be expected to curtail or shut down its plant in order to permit the utility to carry on routine maintenance. Therefore special consideration must be given to this problem in the boiler layout. As indicated by the curve in Fig. 6, there are three boilers per turbine, each having a continuous rated capacity of 200,000 lb of steam per hr and an overload capacity of 230,000 lb per hr. It is of interest to note that, due to the base load which must be carried by the boilers in order to meet the demand for process steam, the loss of one boiler, representing one third of the total boiler capacity, results in an electric-load reduction of more than 50 per cent. Thus it is clear that it is desirable not only to carry some additional reserve boiler capacity in order to offset the slightly lower boiler availabilities with waste fuels, but it is also necessary to install several small boilers rather than a lesser number of large boilers, in order to avoid an excessive load reduction on the turbine generator when one boiler is out of service.

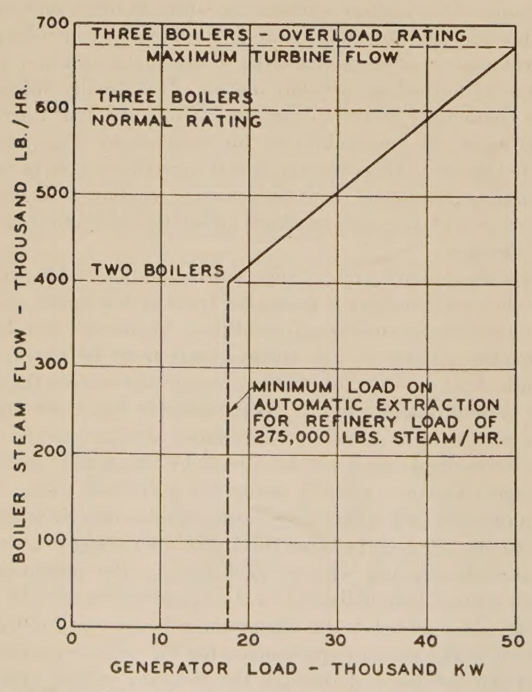


Fig. 6 Boiler Steam-Flow Requirements for Various Turbine-Generator Loads and a Refinery Steam Load of 275,000 Lb Per Hr

Heat Rate for Electric Generation

In Fig. 7 the estimated heat rate per kilowatthour of electric generation has been plotted as a separate item by segregating the proportions of fuel chargeable, respectively, to the generation of process steam and electricity. There are various methods of computing the proper allocation based upon the initial premises which are set up; but for the purpose of establishing the curve in Fig. 7 it was decided to credit the total fuel consumption for both process steam and electricity with the amount of fuel that would be required to generate the contract demand for 275,000 lb per hr of process steam in low-pressure boilers. Then the balance of the total fuel used is all that is chargeable against electric generation. It is fully realized that this method of allocating the fuel for process steam and power is debatable, but a study of other methods of arriving at a solution based upon a breakdown of the various components of the plant cycle led to the conclusion that these also left much to be desired and involved more complication than appeared to be justified for the present objective.

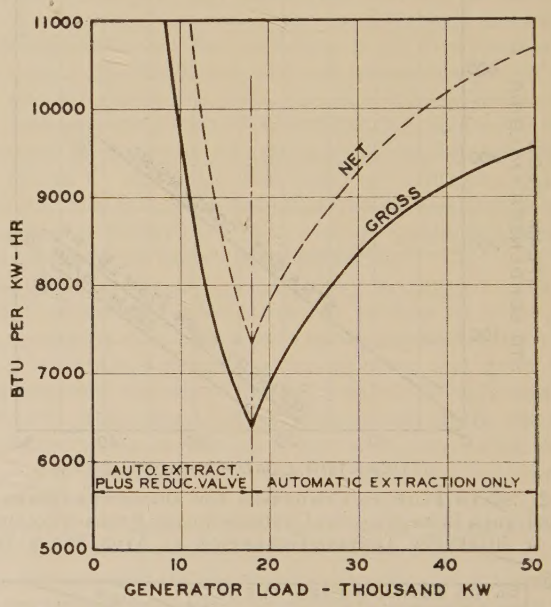


Fig. 7 Heat Rate in Btu per Kwhr of Electric Generation After Deducting Fuel Required for Generation of 275,000 Lb per Hr of Process Steam—Avon Steam Plant

Burning of Waste Fuels

The physical and chemical characteristics of refinery-waste fuels are subject to considerable change from time to time, and the fuel-burning equipment at the plant must be sufficiently flexible to anticipate such changes. The design of the boiler must also be such that fuels which are high in ash and sulphur may be burned with reasonable boiler availability. Some of these liquid-petroleum-base fuels present operating difficulties which are not usually encountered with coal. Therefore the commercial aspects of the contract must be worked out properly so that the extra handling and maintenance costs plus lower boiler availability associated with certain of these waste fuels will not penalize the utility in the cost of electric generation.

The types of fuels burned differ somewhat in each of the three plants under consideration; and from time to time there have also been radical changes both in the type and quality of the fuels delivered to any particular plant due to the changes in the manufacturing programs at the respective refineries. To date the types of fuels which have been burned are given in Table 2. The analyses of these fuels are given in Table 3.

TABLE 2 TYPES OF FUELS BURNED

Avon	Martinez	Oleum
Acid sludge	Recovered oil	Pulverized petroleum coke
Pitch	Cracked asphalt	High-sulphur fuel oil
Heavy fuel oil	Heavy fuel oil	Commercial fuel oil
High-sulphur fuel oil	High-sulphur fuel oil	Natural gas
Commercial fuel oil	Commercial fuel oil	
Natural gas	Natural gas	

TABLE 3 TYPICAL ANALYSES OF CRITICAL FUELS

	Per cent by weight	
	Sulphur	Ash
Acid sludge	2.9–4.6	0.03–0.06
Recovered oil	6.0–9.0	0.01–0.09
Pitch	0.05	0.70
High-sulphur fuel oil	3.0–4.6	0.08–0.14
High-sulphur petroleum coke	4.0–4.2	0.51–0.84

The curves in Fig. 8 show historically the types and quantities of each kind of fuel which have been burned at the Avon plant since it was started up in December, 1940. It is interesting to note that during the war years, large quantities of natural gas were burned as a result of a co-operative effort between the Pacific Gas and Electric Company and the oil refineries on the one hand, and the United States Navy on the other hand, in order to make the maximum amount of fuel oil available for the Navy. Again, during the months of September, October, and November, 1948, it became necessary to burn natural gas in order to keep our plants in operation during the Pacific Coast oil-refinery strike.

Our early experience with the burning of acid sludge resulted in rather low boiler availability owing to external fouling of the boiler heating surfaces; and because of combustion instability, supporting gas or oil pilot fires must be used. However, as a result of research on the part of the oil company, changes were made in its equipment and processes so that the modified acid sludge could be burned with fairly high boiler availability. Due to the high water content, it must not be heated much above 190 F in order to avoid a boilover in the heated storage tanks, and this temperature is satisfactory at the burners.

The problem of burning pitch or cracked asphalt is confined merely to maintaining its temperature at the burners at approximately 400 F in order to obtain a satisfactory viscosity for burning it under the boilers. At ordinary temperatures it is in the solid state, and therefore considerable care must be exercised in seeing that the temperature of this material is never allowed to go too low. This is accomplished by means of steam tracer lines bound to the fuel lines and surrounded by insulation. Other than this, it has proved to be a fairly acceptable fuel which can be burned almost as satisfactorily as the commercial grade of fuel oil derived from the same crude, although it is somewhat higher in ash content and therefore will produce slightly more deposits on the boiler heating surfaces than will the fuel oil.

So-called "recovered oil" is the end product after extracting a large percentage of the sulphuric acid present in acid sludges resulting from the manufacture of lubricating oil. Both acid sludge and recovered oil are highly corrosive, and the products of combustion are very corrosive at temperatures which normally would exist in the air preheater of the boiler. All fuels with high sulphur contents such as acid sludge, recovered oil, high-sulphur fuel oil or coke, and the like, make it necessary to provide for partly by-passing the air preheater so that the exit-gas temperature can be held above the dew point. The effect of the presence of sulphur, or rather of sulphur trioxide, upon the dew point of the exit gases is fairly well understood, and it is necessary, in order to avoid air preheater corrosion, to maintain these temperatures in general conformity with the curves shown in Fig. 9 as the sulphur content in the fuel increases. This results in an appreciable reduction in boiler efficiency which should be taken into account in working out the commercial details of the contract.

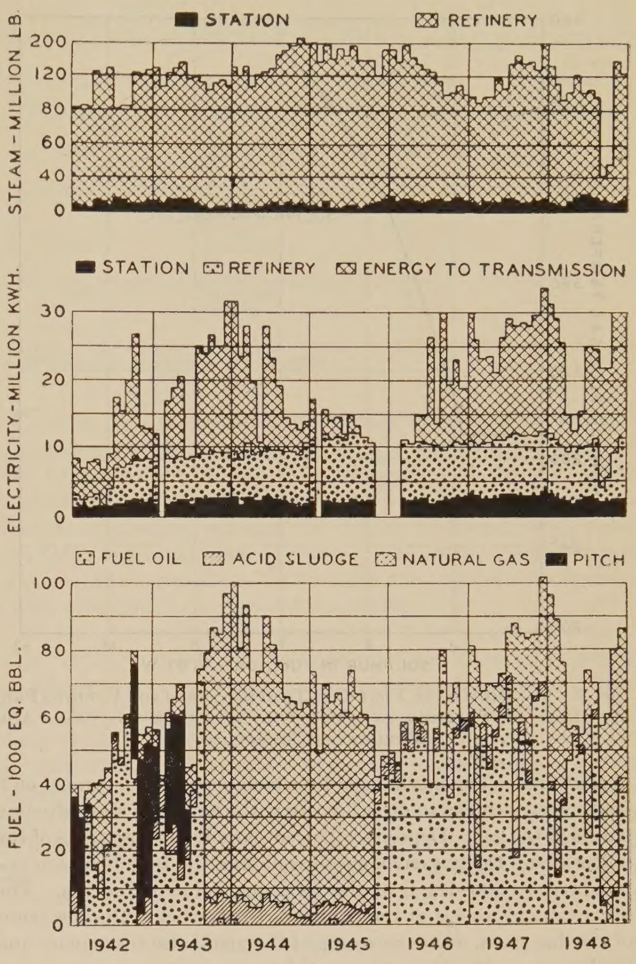


Fig. 8 Historical Chart Showing Process Steam and Electric Energy Delivered to Oil Refinery, and Types of Fuel Burned at Avon Steam Plant

High-Sulphur Fuel Oils and Petroleum Coke

Certain so-called high-sulphur fuel oils and petroleum cokes require special consideration in order to be able to burn them with reasonable boiler availability and maintenance costs. The particular fuel oil referred to here has the sulphur and ash contents shown in Table 3. The degree of external fouling of the boiler heating surfaces appears to be influenced by the combination of both the high sulphur and ash contents. It is not felt that the presence of high sulphur by itself would result in appreciable fouling, but it also has been fairly well demonstrated that, in the presence of a high ash content, the combination will produce a bad fouling problem. In certain temperature zones, a binding action exists which results in maximum deposits on the heating surfaces

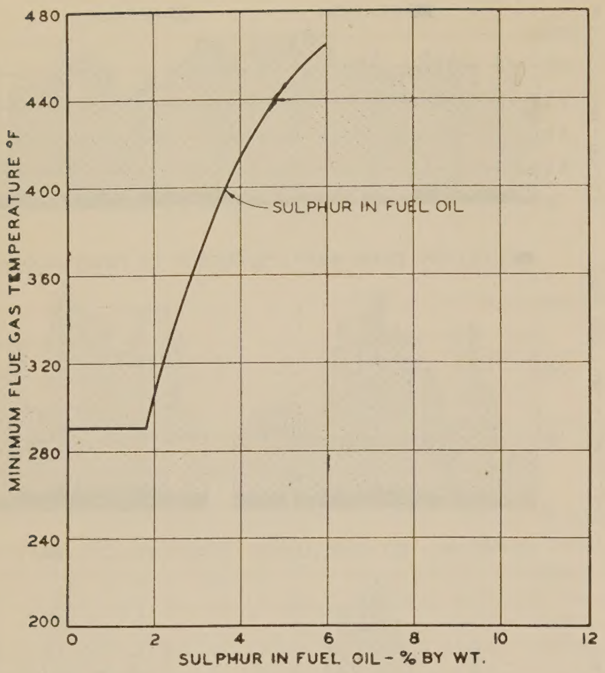


FIG. 9 MINIMUM SAFE FLUE-GAS TEMPERATURES FOR VARIOUS PERCENTAGES OF SULPHUR IN FUEL OIL IN ORDER TO AVOID AIR-PREHEATER CORROSION

in that area. The mechanism by which deposits are formed on a heating surface has to do primarily with the fusion and softening points of the ash. Deposits will occur largely in those parts of the boiler where the gas envelope surrounding the tube is within approximately 200 deg F of the softening point of the ash. The other critical zone is at temperatures near or below the dew point of the flue gases, where corrosion of the metal also takes place and further aggravates the fouling problem.

The characteristic of the ash content in high-sulphur fuel oil produces a fouling problem which is quite different from that usually experienced with coal, owing to the fact that the fusion point of the ash is in the neighborhood of 1600 F. As may be seen from the schematic cross section of the heating surfaces of the boiler shown in Fig. 11, this temperature zone lies in the first bank of superheater tubes. With coals, the fusion point of the ash is frequently well above 2200 F, which fixes the fouling zone in the boiler-generating tubes. In such instances, it is possible to control to a very large extent the whole problem of fouling by going to a low-heat-release furnace, so that the temperature of the gases leaving the furnace are for the most part below the softening point of the ash. The boiler and superheater tubes for coal-firing are widely spaced in order to provide low velocities for the gases of combustion, which appears to be helpful in preventing the fly ash from depositing on the tubes, and also because the mechanical strength of coal slags is usually greater than that from petroleum-base fuels, thereby increasing the tendency to bridge across between tubes.

In connection with the firing of high-sulphur fuel oil and petroleum coke, interesting proof of the sharply defined temperature conditions which control deposits on boiler heating surfaces was obtained by fanning out the first bank of superheater tubes so that the spacing at the bottom was increased from $3^{1}/_{2}$- to $4^{1}/_{2}$-in. centers for $2^{1}/_{2}$-in-OD tubes. This was accomplished simply by inserting 2-in. spacers between the lower ends of the tubes, thus increasing the clear space between tubes from 1 to 2 in. Actually, with only 1 in. of clear space, some of the tubes were much closer than this, or even touching one another due to warpage. The method of accomplishing this is illustrated in Fig. 10, and the effect on the temperature gradient of the flue gases is shown in Fig. 11.

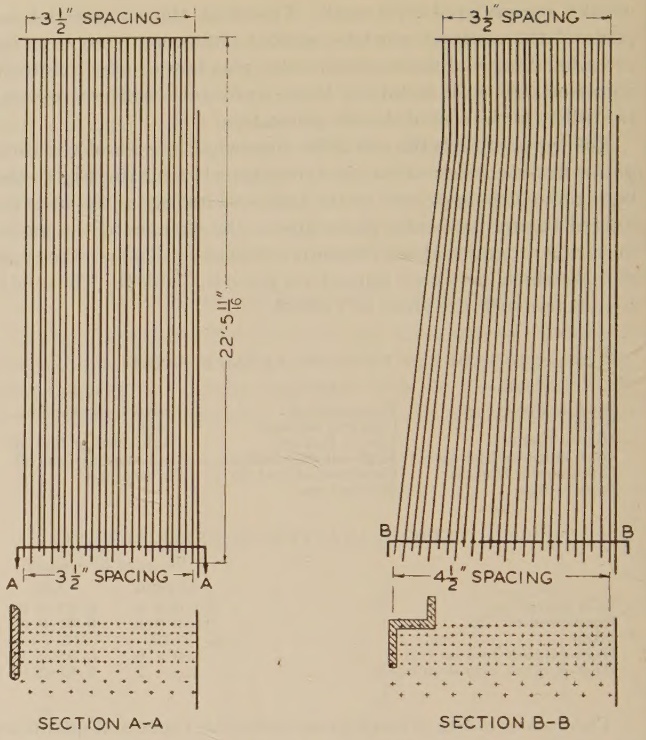


FIG. 10 METHOD OF FANNING-OUT BOILER SUPERHEATER TUBES TO INCREASE CLEAR SPACE BETWEEN TUBES—OLEUM STEAM PLANT

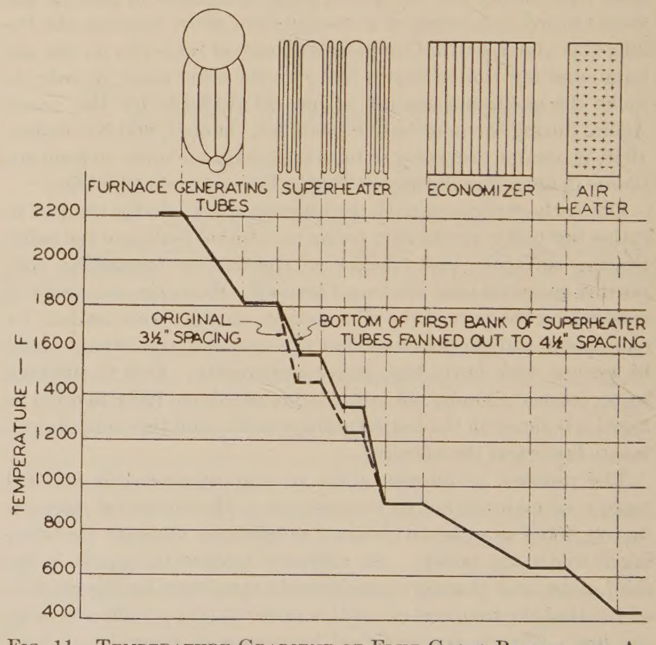


FIG. 11 TEMPERATURE GRADIENT OF FLUE GASES BEFORE AND AFTER FANNING OUT SUPERHEATER TUBES, AS SHOWN IN FIG. 10

Before making the change just mentioned, in the superheater-tube spacing, the fouling zone was concentrated initially in the first bank of the superheater, and as the deposits continued to build up, the temperatures ultimately were raised in the second bank to the point where fouling would commence in that zone. The tubes were so closely spaced that effective soot-blowing was impossible. However, after fanning them out, the amount of fouling in the first bank and the first three rows of the second bank was materially reduced, and it was possible to keep these tubes clean by means of soot-blowing. There was some increase in the amount of deposits in the last few rows of the second bank. It has been concluded that this is due to lack of penetration of the soot blowers. It is anticipated that installation of additional blowers between the second and third superheater banks will remove these deposits.

Increasing the spacing by the method described was considered as an experiment in order to determine the optimum spacing of the superheater tubes. However, such a noticeable improvement in the operating periods between cleanings was obtained at nominal expense that nine of the twelve boilers in the three plants are in the process of being changed in a similar manner. The other three boilers, which are designed for coke-firing as well as fuel oil, are to undergo a more radical change in superheater-tube spacing as a further experiment. The conclusion to be drawn from the foregoing experiment appears to be that soot-blowing can be made effective for these special fuels if the optimum spacing of the superheater tubes is predicted correctly. From this optimum point, the indications are that the effectiveness of the blowers will decrease either for narrower or wider spacing.

Additional problems were encountered in the furnace owing to the particular kind of ash found to be present in either the high-sulphur fuel oil or coke derived from the same crude stock. From the analysis of the ash given in Table 4, it can be seen that appreciable quantities of vanadium and nickel are present, and that the total of NiO, SO_3, V_2O_5, and Na_2O represents nearly 85 per cent of the entire sample. The presence of these compounds introduced a phenomenon of furnace-wall growth which eventually progressed to a point where the expansion spaces at the corners of the furnace walls were completely taken up, thereby setting up very heavy stresses which in turn resulted in serious distortion of the steel members between the buck stays and the furnace walls. This condition was first observed after a period during which approximately 25,000 tons of petroleum coke had been burned.

TABLE 4 ANALYSIS OF ASH IN PETROLEUM COKE

	Substance	Per cent by weight
(1)	SiO_2	7.9
(2)	Al_2O_3	2.6
(3)	TiO_2	0.2
(4)	CaO	1.3
(5)	MgO	3.0
(6)	Fe_2O_3	0.3
(7)	NiO	10.5
(8)	SO_3	20.9
(9)	V_2O_5	29.9
(10)	Na_2O (calculated)	23.4
	Total of items 7-8-9-10	84.7

The furnace walls consist of waterwall tubes which are studded and faced with plastic chrome ore. In the process of combustion, sulphates and vanadates of the alkalies are formed; and these appear to have a strong affinity for the chrome ore and penetrate the pores and cracks which occur during successive cooling and heating of the furnace. Under such temperature reversals these compounds pass back and forth from the solid to the liquid state, thereby penetrating the chrome ore with resultant growth in volume. The coefficient of expansion of the slag is greater than that of the chrome ore, and the combination produces a very dense mass which brings about a progressive growth of the entire wall. The resultant mechanical forces which are set up are sufficient to force the water-wall tubes to spread further apart until all the expansion space in the corners of the furnace walls has been taken up. This condition has been overcome by the addition of more tie bars for holding the waterwall tubes in place, and the substitution of a different type of refractory facing for the tubes.

In Table 5 is shown a rough correlation between the proportion of critical fuels burned and the average number of boiler shutdowns per year for external washing of the heating surfaces. It can be noted that, during 1945, on account of the war, much smaller percentages of critical fuels were burned, and the average number of cleaning outages was the lowest during the 4 years shown in the tabulation. However, due to the improvements which are being made in overcoming the problem of deposits, boiler availabilities increased during 1948.

TABLE 5 PROPORTION OF CRITICAL FUELS BURNED, COMPARED WITH AVERAGE NUMBER OF BOILER SHUTDOWNS PER YEAR FOR EXTERNAL CLEANING OF HEATING SURFACES

Year	Station	Proportion of critical fuels burned, per cent of total	Average number of cleaning outages per year per boiler
1948	Avon	45	2.7
	Martinez	69	2.7
	Oleum	59	1.7
1947	Avon	66	4.0
	Martinez	73	3.0
	Oleum	78	1.7
1946	Avon	92	3.0
	Martinez	70	2.7
	Oleum	73	2.0
1945	Avon	18	1.7
	Martinez	17	1.7
	Oleum	14	1.2

NOTE: The number of cleaning outages in the last column includes the regular annual overhaul and inspection.

This problem of burning special types of fuels has been treated at some length because it has to do with one of the major points of difference between a conventional plant and the semi-industrial type which is the subject of this paper. On the other hand, the data which have been presented clearly show that, although the problem should not be underestimated in entering upon a joint undertaking of this nature, waste fuels can be burned with reasonable boiler availability, and that operating experience has brought to light the possibility of boiler-design modifications and improved soot-blowing techniques, which should bring these availabilities still closer to what can be obtained with commercial grades of fuel.

RAW-WATER SUPPLY

The large quantities of raw water required for the process evaporators are supplied from a slough in the San Joaquin River about 10 miles north of the Avon steam plant. However, due to a combination of tidal conditions and seasonal variations in river flow, the salt water from San Francisco Bay tends to back up in the river and the salinity of the water becomes prohibitive at certain seasons of the year. Shortly after the Avon station was started up, an alternate supply became available from the Contra Costa Canal which is an irrigation project of the United States Bureau of Reclamation. The source of this canal is located at a higher point up the San Joaquin River, but conditions are such that, at certain seasons of the year, the salinity from Mallard Slough is less than that which prevails in the Contra Costa Canal, and it has been found desirable to switch from one source to the other during the year, as indicated in Fig. 12. The chemical conditioning of this raw water for evaporation involves a formidable treating plant which is illustrated in Fig. 13. The average analyses of the raw water are given for both Mallard Slough and the Contra Costa Canal and the analyses at various points in the treating cycle are also given.

The problem of preparing the raw water for evaporation without carry-over and consequent contamination of the entire system is of major importance. However, it is felt that this is a subject which does not stand out as peculiar to this type of plant, except with respect to the very large quantities of water to be handled. Therefore the matter will not be elaborated here in such detail as has been the case with the special problems directly related to this type of station.

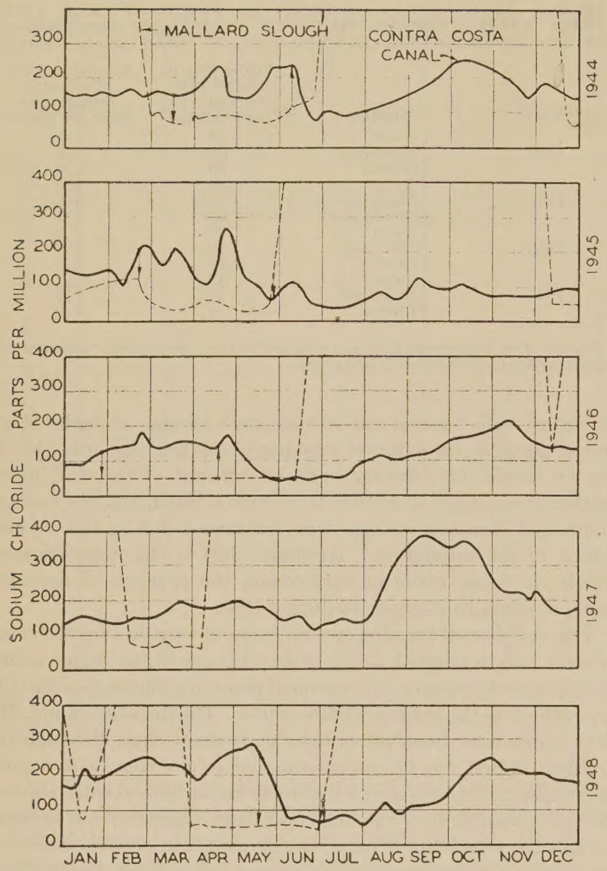


Fig. 12 Historical Record of Salinity of Raw-Water Sources for Avon and Martinez Steam Plants

Vertical arrows indicate time of change-over from one source of supply to the other.)

Conclusion

An attempt has been made to point out and evaluate the special problems which enter into the design and operation of a plant of this type. The advantages that are inherent in such a joint undertaking should be obvious, but it is possible to wipe out a large proportion of these benefits by the failure to give proper weight to the various factors involved. It is therefore hoped that the somewhat detailed analysis of the more important problems which have been set forth in this paper will be helpful in developing a sound basis for making a realistic appraisal of any such similar undertaking which might come up for consideration in the future.

Discussion

F. S. Cummings.[3] This paper is of particular interest on two points:

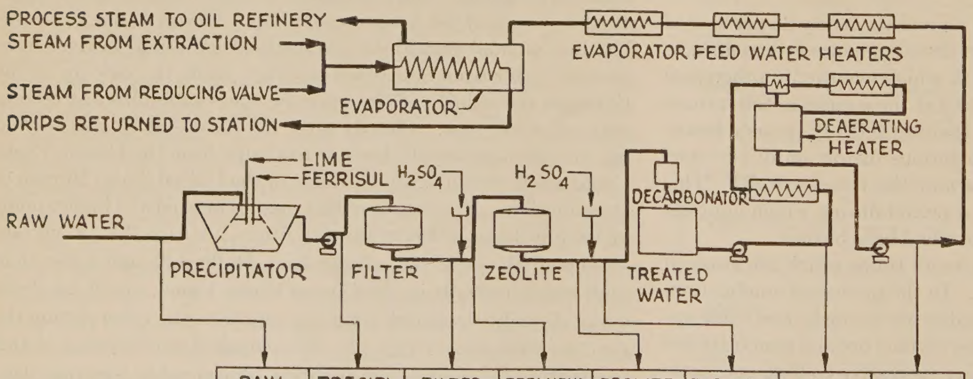
1 It gives operating results of a modern high-pressure public-utility generating plant, operating in conjunction with a large industrial plant, requiring both electric energy and large quantities of moderately high-pressure steam for process work.

2 It covers what is believed to be the first published information on the problems of burning, under modern high-pressure boilers, practically all of the waste products from a modern oil refinery.

In the last 25 or 30 years there have been many industrial plants built to supply process steam and electric energy for plant uses. Sometimes these plants operate as independent units, and sometimes they operate in parallel with a public-utility system to better take care of any unbalance in electric and steam requirements of the industry. However, there have been relatively few public-utility plants built for the express purpose not only of supplying electric energy to the utility distribution system, but

---

[3] Vice-President and Chief Engineer, C. C. Moore & Company, Engineers, San Francisco, Calif. Mem. ASME.



		RAW WATER	PRECIPITATOR EFFLUENT	FILTER BEFORE ACID FEED	EFFLUENT AFTER ACID FEED	ZEOLITE BEFORE ACID FEED	EFFLUENT AFTER ACID FEED	AFTER DECARB. AND PO_4	DEAERATOR EFFLUENT
MALLARD SLOUGH	TURBIDITY- PPM	18-71	0-10	0	0	0	$(CO_2$	-Tr. - 5	0)
	NaCl - "	20-140	20-140	20-140	20-140	20-140	20-140	20-140	20-140
	TOTAL H - "	25-45		42-46		0	0	0	0
	P ALK. - "	0	25-30	25-30	0	0	0	0-10	15-25
	M ALK. - "	35-45	45-50	40-45	12-17	14-19	10-15	14-19	20-35
	pH	7.3	10.3-10.5	10.5	7.1-7.3	8.0-9.0	6.1-6.5	8.0-9.0	9.3-9.8
	TOTAL SOLIDS	150-350	150-350	140-300	140-300	140-300	140-300	140-300	140-300
CONTRA COSTA CANAL	TURBIDITY PPM	0-80	0-10	0	0	0	$(CO_2$	-Tr. -3	0)
	NaCl - "	55-380	55-380	55-380	55-380	55-380	55-380	55-380	55-380
	TOTAL H - "	140-350		90-142		0	0	0	0
	P ALK. - "	0-0.3	55-30	52-25	0	0-7	0	0-8	20-15
	M ALK. - "	115-190	100-45	95-35	30-15	28-13	20-10	30-15	35-20
	pH	8.1-8.3	10.3-10.5	10.3-10.5	7.3-7.9	7.5-8.3	6.5-7.8	8.1-8.3	9.0-9.3
	TOTAL SOLIDS	150-570	150-570	150-550	150-550	150-550	150-550	150-550	150-550

Fig. 13 Flow Cycle and Chemical-Treating Cycle for Raw-Water Supply to Process Evaporators at Avon and Martinez Steam Plants

also to supply electric energy and steam for an adjoining industrial plant. Further, there have been still fewer such plants built to operate at high pressures and temperatures.

As pointed out in the paper, both the utility and the industry can gain much by such a tie. The advantages to be gained are so many that future planners should consider them seriously. Probably one of the more important reasons for a utility being reluctant to tie in with an industrial plant has been the fear on the part of the utility that it would lose some of its flexibility in the generation of electric power which of course is its basic reason for existence. The plants discussed in the paper have been in operation for a number of years, and the operation has given ample proof that there need be no serious difficulties.

Until perhaps the last 15 or 20 years, fuel oil was considered to be perhaps the most ideal of fuels. Compared to the use of coal and other fuels, boiler outage was relatively low, maintenance was also relatively low, the elimination of expensive storage, handling, and burning equipment meant lower first cost, and operation was generally more flexible. All of this has changed in more recent years. The change in quality of fuel oil available during the war years, and changes in refining methods, have created many new problems for the boiler designer and operator—problems which are not always solved by referring to experience in burning coal. Years ago oil-fired boilers were relatively small, of comparatively low pressure, and ran at moderate ratings. Central-station boilers today are of large capacity, high pressure and temperature, and operate at high ratings. These factors have increased the problems confronting the designer and operator.

On the West Coast the quality of fuel oil also has deteriorated because of the decreasing supply of crude, which has necessitated an increased use of crudes which are relatively high in impurities. These impurities are sulphur and sometimes small quantities of rare metal oxides which have a peculiar action on the ash which is in the fuel. While fuel oils themselves many times have these troublesome characteristics, the various types of waste fuels, burned in the three plants under discussion, produce aggravated difficulties because of the higher percentage of impurities.

It is most interesting to note that boiler and plant design has been such that even the worst of the waste fuels have been handled with some considerable success. The operating experience gained has been most valuable in the design and operation of new units; it is by studying the operating difficulties encountered that future plants can be built with increased reliability equal to the best of modern coal-burning plants.

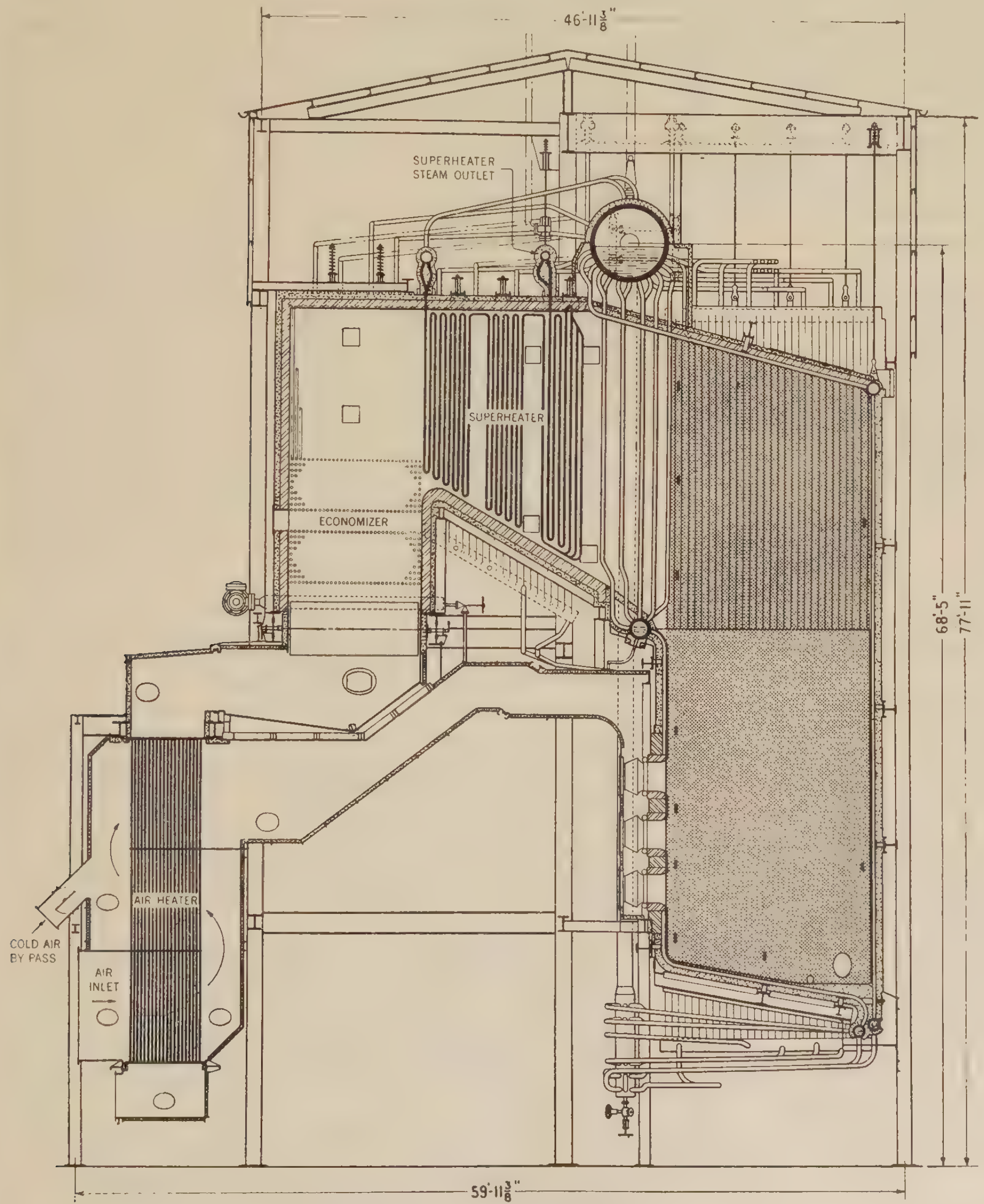


Fig. 14 Cross Section Through Steam-Generating Unit as Installed at Oleum, Avon, and Martinez Stations

F. X. Gilg.[4] It may be of interest to show a cross section through the steam-generating units which were installed as part of the Oleum, Avon, and Martinez projects, described by the author.

Fig. 14 herewith shows a cross section through the boilers which are radiant-type single-pass gas flow, designed to generate 200,000 lb of steam per hr continuously at 1450 psi and 950 F at the outlet of the superheater. Final steam temperature is controlled by means of a gas by-pass around the superheater at one side of the boiler, the quantity of gas by-passed being controlled by dampers located in the cooler zone under the economizer.

---

[4] Application Engineer, The Babcock & Wilcox Company, New York, N. Y. Mem. ASME.

The furnaces are completely water-cooled and designed to accommodate oil, gas, or pulverized fuel.

The economizer is of the continuous-tube type and extends across the full width of the setting including the superheater by-pass area.

A cold-air by-pass is provided in the tubular air heater so that the tube-metal temperature can be raised to minimize corrosion when burning high-sulphur fuels.

The boilers are installed outdoors with just a protecting roof over them.

As pointed out by the author, experience gained on these units burning refinery-waste fuels has been helpful in designing later units so that the superheater surfaces do not foul up as quickly and can be cleaned by soot blowers in service. As can be seen from the temperatures in Fig. 11 of the paper, it is more complicated than simply providing more space between superheater tubes, because the absorption decreases with wider spacing, resulting in higher gas-temperature levels in the subsequent banks of superheater tubes and the possibility of transferring the troublesome area. It is to be noted that after fanning the lower ends of the first-bank superheater tubes so that the clear space was 2 in. instead of 1 in. between tubes, the temperature of the gases entering the second bank of superheater tubes increased from 1450 F to 1580 F. However, indications are that the wider spacing resulted in less plugging of the first bank of superheater tubes, which was so troublesome from the beginning.

F. G. Philo.[5] Experience obtained in burning natural gas and high-impurity fuel oils in large quantities at the Long Beach Steam Station, through evolution, has led to a number of boiler-design, operating, and maintenance betterments which are incorporated in the boiler installation of the Redondo Steam Station which was placed in service late in February, 1948. In connection with the present paper it may be helpful to mention several of these improvements. The more important changes are as follows:

1 Elimination of all gas baffles.

2 Graded spacing of furnace screen, superheater and economizer tubes.

3 Watertight acid-resistant sloping floor and hopper beneath superheaters, economizers, and air preheaters to facilitate water washing.

4 Downflow of gases through economizer and air preheater.

5 Permanent provision to collect and conduct wash water with dissolved and other solids to a point of convenient disposal.

In a general way, with high-impurity fuel oils, including rather high sulphur content, such as Pacific Coast 400 or Bunker C, the boiler, superheater, economizer, and air-preheater surfaces operate within a very wide range and number of gradations of skin temperatures (on the surfaces of deposits especially) and function quite effectively as fractionating condensers.

Most of the impurities enter the boiler passages in the vapor or gas phase and tend to deposit throughout the heat-transfer system wherever temperature and moisture conditions are favorable for deposit. The situation to a large degree may be likened to the changing nature of deposits often found in successive stages of a steam turbine due to carry-over of solids from the boiler.

Referring to the accompanying sectional outline of one of the Redondo boilers, Fig. 15, the changes previously itemized will be noted as follows:

TABLE 6 TUBE SPACINGS

Location	No. rows deep	Tube OD, in.	Space between tubes, in.	
			Across	Deep
Furnace screen....................	3	4	14	8
Furnace screen....................	2	4	5	8
Secondary superheater.............	3	$2\frac{1}{2}$	$4\frac{1}{2}$	$2\frac{1}{2}$
Secondary superheater.............	5	$2\frac{3}{4}$	$4\frac{1}{4}$	$2\frac{1}{4}$
Primary 2nd section superheater....	8	$2\frac{1}{2}$	1	$1\frac{1}{2}$
Primary 1st section superheater.....	10	$2\frac{1}{2}$	1	$1\frac{1}{2}$
Economizer 2nd section............	21	$2\frac{1}{2}$	2	$1\frac{1}{2}$
Economizer 1st section............	20	$2\frac{1}{2}$	2	$1\frac{1}{2}$

1 Elimination of all gas baffles facilitates water washing; certain areas may be washed both from front and back.

2 The lateral and "front to back" tube spacings are as given in Table 6 herewith.

Gradation of cross spacing between tubes has prevented appreciable increase of draft loss due to fouling.

Although, at the time of installation, the boilers were equipped with the latest design of retractable mass-action soot blowers, deposits do adhere to the various heating surfaces and cause a gradual but slow reduction in the range of superheater control.

The greatest deposits occur on the surfaces of the economizer tubes. This particular deposit is rather loose and soft and, fortunately, is highly soluble and easily removed with a generous stream of water slushed over the top row of tubes in each of the two tube banks of the economizer. This is done with a very low fire and can be completed in a period of about 2 hr "off-line" at full boiler pressure.

3-4-5-Watertight, acid-resisting floors and soot hoppers, downflow of the gases and wash water in the economizers and air preheaters, and washing toward the cold end of the economizers and the air preheaters, especially in the case of the air preheater, combine to wash deposits out of the preheater instead of through the air preheater.

To those unacquainted with the problem of deposits from relatively high-ash and sulphur fuel oils, it may be of interest to know of the quite effective cleaning job that may be accomplished by the occasional use of natural gas for several days, as a cleaning agent for surfaces fouled by noncombustible residues from fuel oil. Where limited amounts of natural gas are available in oil-burning plants, it has been found good practice to rotate or alternate the firing of gas and oil. The reasons for the cleaning effect of gas are still undetermined.

Operation with regenerative air preheaters to date has been encouraging as most of the troubles previously experienced with the older tubular air-preheater deposits have not been encountered. It is felt that a great deal of the corrosion and deposits difficulties previously experienced with the tubular air preheaters was chargeable to uneven distribution of air and gas flow around and through the individual tubes. Scott Jensen[6] has surveyed the pattern of gas temperatures from the individual tubes of large preheaters at the Long Beach Steam Station which were installed in 1928–1930. Corrections of local flow rates in the preheaters have raised the lower local gas temperatures sufficiently to mitigate corrosion greatly and also plugging by deposits on the cold ends of the air preheater tubes.

The writer feels that the problem of oil-ash residues has not received its warranted study by the boiler and soot-blower manufacturers. Intensive study of this important problem is suggested as it is reasonable to expect increasing troubles with higher concentrations of oil-impurity residues anticipated from fuel oils produced by improved refinery processes of the future.

W. F. Ryan.[7] The advantages of co-operation between process

---

[5] Chief Mechanical Engineer, Southern California Edison Company, Los Angeles, Calif. Mem. ASME.

[6] Steam Plant Engineer, Southern California Edison Company, Los Angeles, Calif. Mem. ASME.

[7] Engineering Manager, Stone & Webster Engineering Corporation, Boston, Mass. Fellow ASME.

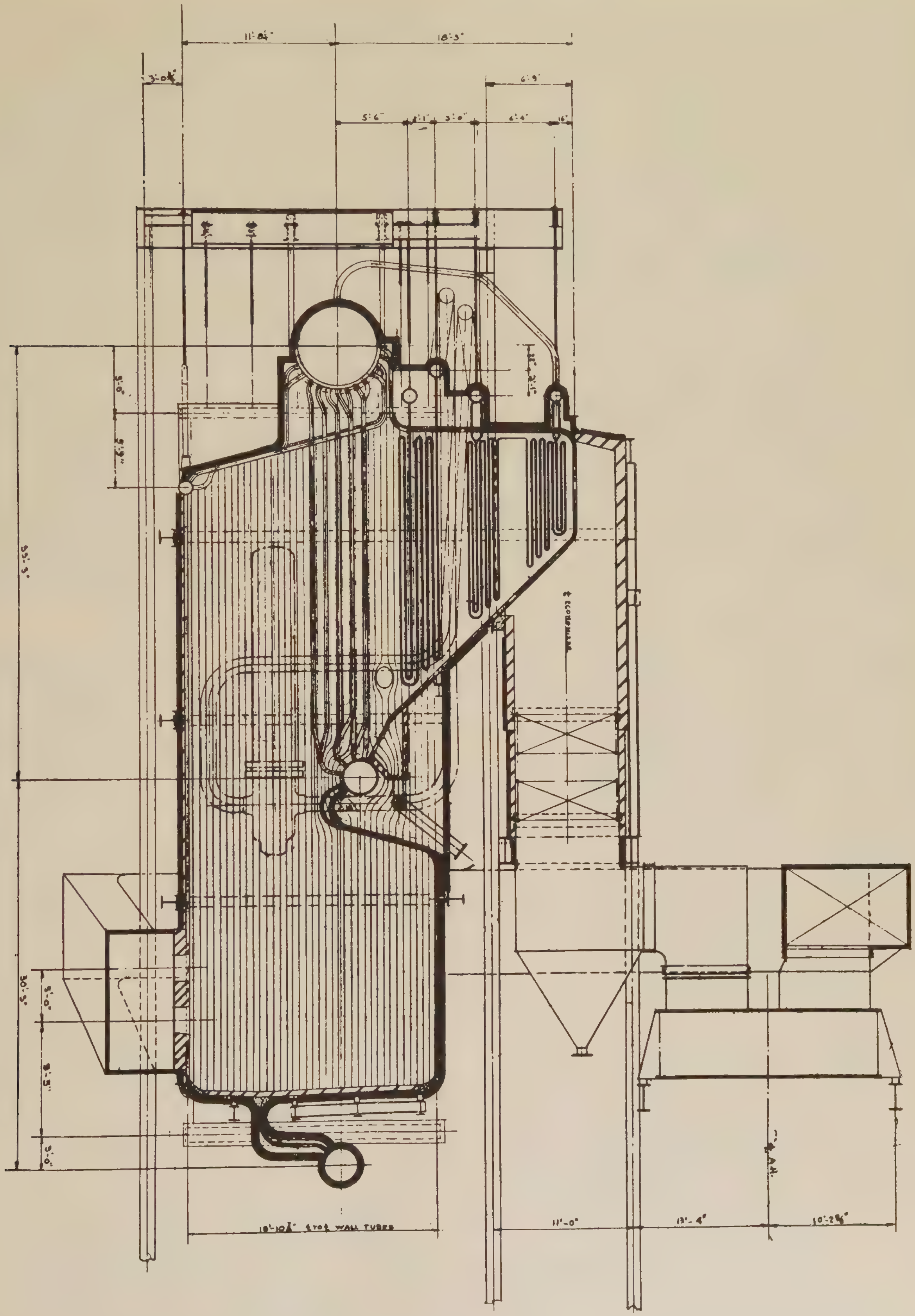


Fig. 15 Cross Section Through Steam Generator at Redondo Station of the Southern California Edison Company
(Maximum capacity 400,000 lb of steam per hr; design pressure 1000 psi; total steam temperature 900 F.)

industries and public utilities for wholesale generation of electric power have never been better stated than by the author in this paper. If these advantages were more widely comprehended, an appreciable portion of steam-generated power in this country might be derived from plants like those described in the paper, with benefit to the national economy from availability of lower-cost electric power, lower-cost process steam, and conservation of our fuel resources. As the author points out, the advantages of the setup should be equitably divided between the utility and the manufacturer. An equitable arrangement is not always easy to arrive at, and it is even less easy to convince both parties that the arrangement is equally advantageous to each. Nevertheless, meticulous and patient appraisal of each detail will be well-rewarded whenever the basic conditions are favorable for such an enterprise.

When these plants were built, the pressures and temperatures selected were probably regarded as the maxima commercially practicable. There should be no obstacle, in the design of future plants, to the use of higher pressures and temperatures. Certainly, if higher pressures and temperatures can be justified for the straight-condensing units in the normal utility plant, there is much greater justification in plants such as these when a substantial portion of the output is generated with a back pressure in excess of 200 psi. Raising the pressure from 1500 to 2500 psi, and the temperature from 940 F to 1050 F increases the energy potential about 8.5 per cent, with 1.5 in. Hg back pressure, but the corresponding increase, with a back pressure of 240 psi, is over 31 per cent. The yield of "by-product" power is correspondingly increased.

Further increase in the yield of by-product power could be effected by interposing a desuperheating heater in the extracted-steam connection between the turbine and the evaporators. The author does not indicate the temperature of the extracted steam, but with a throttle temperature of 940 F, the amount of superheat at 200–240 psi must be substantial. It has been shown by Cross and Wells[8] that the use of the feedwater to desuperheat this steam before it enters the evaporator provides an increase in the yield of by-product out of all proportion to the cost of the simple equipment required. For the steam conditions at the Pacific Gas & Electric Company plants, the increase would be about 8.0 per cent.

With 2500 psi initial pressure, reheat would be desirable for the condensing part of the cycle. With this reheat taking place at about 470 psi, the combined effect of higher pressure, reheat, and the interposition of a desuperheating heater in the extraction line would increase the yield of by-product power by more than 48 per cent. This larger noncondensing operation probably would justify the use of turbine-generators of 60,000 to 80,000 kw, which might be more suitable for large power systems such as that of P.G. & E.

These thermodynamic considerations are well-known, and relatively unimportant in comparison with the business aspects of this kind of undertaking, and the practical operating results which are so ably reported in this paper. We are greatly indebted to the author and to the Pacific Gas and Electric Company for making these data public. It is hoped that this report may inspire other utilities and other process industries to develop like opportunities to the advantage of themselves, and to the benefit of the public which ultimately profits both from lower costs of power and lower costs of processed products.

H. G. Thielscher.[9] A significant fact is the foresight shown by the designers of the station described in selecting a steam pressure of 1500 psi and 940 F 12 years ago when that temperature was not at all common, even in central-station practice. This selection of steam conditions leaves the utility in an attractive position today should it ever become necessary to operate these plants as electric-generating stations only.

Fig. 3 of the paper indicates a throttle flow of 680,000 lb per hr for 50,000-kw turbine load and a refinery flow of 275,000 lb of steam per hr. For the steam conditions selected and with a straight regenerative cycle and 4 stages of extraction, we might expect a throttle flow of 450,000 lb for 50,000-kw turbine load. The difference between this figure and the 680,000 lb of steam provided, or 230,000 lb per hr, represents a considerable additional investment in the cost of the boiler plant.

Other major items which make the plant cost more than a conventional electric-generating station are the turbine-generator, process-steam evaporators, process-steam piping, and water-purification plant.

No doubt these factors were taken into consideration when figuring the charge for process steam and electric energy. As an offset to the additional fixed charges imposed by these items, it is to be noted that the heat rate per net kilowatthour generated averages much less than that of a conventional plant.

The oil industry generally practices the policy of writing off its investment in new equipment in a few years compared to utility practice of, say, 30 years. It would be interesting to know whether a quick write-off of the extra investment involved in this plant, compared to a conventional generating plant, was figured in arriving at the charges to be made for the products supplied to the refinery.

P. H. Woods.[10] It has been both a pleasure and a privilege to have participated from its inception in the development of the arrangement for supply of steam and electric power by the Pacific Gas and Electric Company to the oil companies as described in this paper. The author has outlined the principal features of the contract and of the technical problems encountered in the operation of the steam plants. Therefore the writer will limit his comments to the arrangement as seen from the oil-company point of view.

In the $8^1/_2$ years that have elapsed since utilities were first supplied to us by the Avon Steam Plant, Tide Water Associated Oil Company has had little reason to regret entering into the contract. The oil company is in the business of making petroleum products, and we have been quite happy to leave generation of power in the hands of the utility—that is, the business they are in.

At the time the present arrangement was first suggested, Associated was in the midst of a major plant expansion which included enlargement of our steam-generating facilities. We had, in fact, entered into a contract for construction of a new 850-psi-pressure boiler plant and were seriously considering installation of electric-generating facilities as well. Fortunately, as we see it now, the suggestion came early enough in our program to permit cancellation of our boiler-construction contract and to switch to the present setup.

Before the Avon Steam Plant was built, Associated generated all of its own steam, partly at 150 psi pressure, and partly at 250 psi, and purchased electric energy from the utility through a single substation at 2400 volts. Now we take delivery of steam at 165 psi pressure with some superheat, and receive electric power through three substations. The old 150-psi boilers have been out of service for some years, as the contract maximum on steam gives us a comfortable margin over actual demand. The arrangement has encouraged us to electrify new construction for the sake of economy in operating costs, and in many cases has enabled us to replace old steam-driven units with electric drives at substantial savings.

---

[8] "An Improved System in the Application of Noncondensing or Extraction Turbines," by H. W. Cross and E. S. Wells, Jr., Trans. ASME, vol. 62, 1940, pp. 37–40.

[9] Resident Engineer, Combustion Engineering-Superheater, Inc., New York, N. Y. Mem. ASME.

[10] Chief Engineer, Tide Water Associated Oil Company, Associated, Calif. Mem. ASME.

Economies have accrued in another direction as well. Formerly we burned sludges and other wastes in a disposal furnace. These noncommercial products are now turned over to the utility in part payment for energy. Some conditioning of these fuels is necessary, but generally this can be accomplished by minor modifications in the manufacturing process. Refinery operating personnel works in close co-operation with steam-plant operators as to characteristics of this type of fuel.

Reliability of supply of electric energy to the refinery has been improved materially. Our second and third substations are fed directly from the steam plant 12-kv bus, and interruptions to this supply are rarities. This factor is important to us, as process units built since 1940 are much more complex than older units, therefore more difficult to get back on stream after a power interruption, and shutdown time is expensive.

In normal steam-plant operation, three evaporators are on-stream to supply the refinery steam load. Fouling requires that they be taken down, one at a time, for periodic cleaning. At such times, when availability is down one third, Associated supplies the difference from its 250-psi system. This second refinery steam system is necessary for process reasons and was in existence prior to the time the steam-plant studies were begun. For several reasons, both process wise and economic, no attempt was made to incorporate a 250-psi steam source into the steam-plant design.

AUTHOR'S CLOSURE

The author is very much indebted to all of those who have presented written discussions. It is felt that all of the remarks are timely and constructive, and have added materially to the value of this paper. It is interesting to note the great amount of interest displayed in the subject of external fouling of boiler heating surfaces.

The observation by Mr. Cummings, to the effect that the problems encountered in burning present-day fuel oil are not always solved by referring to experience in burning coal, has been well substantiated by the experiences of plants on the eastern seaboard which have recently changed from coal to fuel-oil firing. Such plants, originally designed for firing with coal, have changed to fuel oil because of the availability of the latter fuel at competitive prices. In some instances the resulting boiler availabilities have been reduced below that which is obtained on the Pacific Coast with the similar types of fuel oil. This is partly due to the fact that, when firing with fuel oil, deposits tend to build up in the first superheater bank which, in a boiler designed for coal firing, sometimes consists of as many as 12 or 14 rows of tubes, thus making it impossible to obtain effective soot-blower penetration. As has already been pointed out, slag deposits do not usually occur in this zone when firing coal due to the usually much higher fusion point of the ash. Effective results from soot blowing cannot be obtained on superheater banks deeper than about six or eight rows of tubes. Satisfactory penetration is obtained by blowing from both the front and back of such banks with mass type blowers.

With reference to the last paragraph of Mr. Gilg's discussion, it is important to emphasize that the one inch clear space between tubes in the first superheater bank cannot consistently be maintained due to a certain amount of warpage which takes place in this temperature zone, thereby producing a very serious obstruction to the path of the gases and making it almost impossible to obtain effective soot-blower penetration. A cross section of one of the boilers is included as Fig. 14 in Mr. Gilg's discussion, and by comparing this with Section AA or BB in Fig. 10 representing the lateral spacing of the tubes in the first superheater banks, it will be noted that there are three rows of relatively wide-spaced tubes at the front of this bank. This arrangement provides a clear space of 3 in. between these tubes, but tends to reduce the effective penetration of the soot blowers through the more closely spaced tubes because of the interference of these first three rows, which also have the effect of keeping the soot blowers further away from the close-spaced rows. Thus the wide-spaced rows, particularly in view of the fact that they are staggered, appear to do more harm than good from the standpoint of fouling of the heating surfaces.

As an experiment on one boiler, the first two staggered rows of wide-spaced tubes have been moved back into line with the third row, thus having the effect of adding one more close-spaced row in place of the three wide-spaced rows, and making it possible to move the soot blowers a distance of approximately 10 in. closer to the close-spaced rows in that bank. This is very important from the standpoint of effective soot-blower penetration, and there is every indication that this modified tube arrangement, plus 2 in. clear space between the tubes and the proper installation of mass blowers, will make it possible to keep this first superheater bank clean for considerably longer periods of time. It has also been demonstrated that the effectiveness of mass-type blowers can be greatly increased by the addition of specific quantities of water. This applies to either air or steam blowers, and results in far less temperature shock than occurs when 100 per cent water washing is resorted to under load. A few plants have actually adopted the latter practice at intervals of, say, 30 days to augment regular soot blowing.

Mr. Philo's discussion is very pertinent and brings out some of the design modifications which have been adopted almost universally on recent Pacific Coast installations. The provisions cited in Item 3 of Mr. Philo's discussion, to facilitate water washing of the boiler while it is off the line, tend to produce a better job of cleaning with less boiler outage time. In connection with Item 2, it has already been emphasized by the author that increased spacing between superheater tubes to some figure greater than 1 in., particularly in the first bank, is highly desirable. However, it does not follow that, by progressively increasing this spacing, we shall obtain correspondingly improved results from the standpoint of less deposits. Operating experience during the past year suggests that there is an optimum spacing which will obviate abnormal restriction of the gases due to slag deposits and at the same time provide for maximum effectiveness of the soot blowers. It is fairly certain that this optimum clear spacing is somewhere between 2 and 3 in. for the superheater banks, and approximately 2 in. for the economizer section. For oil fuel, it is doubtful whether spacing as little as one inch can be justified at any point in either the superheater or economizer sections.

The investigation conducted by Scott Jensen and referred to by Mr. Philo in connection with the survey of the pattern of gas temperatures leaving individual tubes of large tubular air preheaters, represents a valuable contribution to the problem. It provides a basis for the installation of baffles so that proper gas-flow distribution is obtained throughout the preheater, thereby eliminating localized low-temperature zones which have a major influence on both corrosion and plugging by deposits at the cold end of the preheater.

With reference to Mr. Ryan's comment on the gains which can be realized by desuperheating the extraction steam before it enters the evaporator, it should be stated that this has been accomplished, at least in part, by means of a steam-to-steam heat exchanger whereby the superheat in the high-pressure turbine exhaust is removed by adding superheat to the evaporator vapor. The refineries require 50 F superheat in the process steam at the point of delivery. In the last paragraph of Mr. Ryan's discussion he has emphasized the importance of the business aspects of this kind of undertaking. This is indeed true because it is absolutely essential to make proper allowance in the financial setup for the various technical factors which enter into the operation of such an undertaking. However, it cannot be too strongly emphasized that an intimate knowledge of the practical operating problems and related technical factors are the essential prerequisites to the conclusion of a satisfactory business arrangement between the two parties.

In connection with Mr. Thielscher's remarks regarding the increased investment in boiler plant over what would be required for a conventional plant, it should be pointed out that additional boiler capacity is required to generate the process steam which is sold to the oil refineries. The working agreement, of course, provides that the revenue from such steam sales will be adequate to pay for this incremental capacity, and it should also be pointed out that the additional high-pressure steam generates by-product power in the high-pressure turbine at extremely high efficiencies. In reply to Mr. Thielscher's question in the last paragraph of his remarks, it should be stated that all three of the contracts with the oil refineries were written for a period of 20 years with provisions for automatic extension beyond that period unless twelve months' written notice is given by either party stating its desire to terminate the agreement.

It is gratifying that Mr. Woods has contributed to the discussion by giving the point of view of the oil refinery, setting forth the advantages which have accrued to them as a result of this undertaking. We should like to add that the day-to-day working relations between the respective operating personnel of the utility and the refinery have been highly satisfactory and consistently take place in the spirit of fine co-operation.

Steam-Electric Power Expansion in Southern California

By W. L. CHADWICK,[1] LOS ANGELES, CALIF.

This paper presents the background of the postwar expansion of steam power in Southern California, the requirements of the area served, a brief discussion of some of the war and postwar influences bearing upon power needs, the disposition of the new generation, the design trends in the area, and the influences bearing thereon. A description is given of the major features of the Redondo Steam Station of the Southern California Edison Company.

INTRODUCTION

FOLLOWING the late war and the necessity of restricting to the maximum the manufacture and installation of electric power-generating equipment of every kind and also the purposeful loading of all existing reserve generation with war production, it has been a common experience throughout the country to have postwar power loads increase more rapidly than new generation capacity could be provided. Perhaps, however, the situation in California is sufficiently different to be noteworthy and of interest.

A companion paper[2] treats of the northern part of the state; this paper will treat of the southern part. California may be said to consist economically of three metropolitan and industrial areas, and a large agricultural hinterland in which there are also some industrial and residential centers. The metropolitan areas of course are those around San Francisco Bay in Northern California, and those around Los Angeles and San Diego in Southern California. Although geographically, Southern California includes only ten of the state's 58 counties and only 37 per cent of the area of the state, it contains more than one half of the state's population and utilizes about one half of the electric power. Another fact which has a large bearing on power resources, particularly hydraulic power, is that the Southern California area possesses only about 1 per cent of the state's water resources.

Because fuel was then scarce and steam power generation was inefficient, the first electric power developed in the state was hydraulic. However, during the first decade of power development, a severe drought so reduced the water supply that some systems found it necessary to move boilers and engines to hydro plants to meet the deficiencies in the hydro output. Since that time, steam power has been an intimate part of power supply, but being always more costly to operate than hydro, it has been used only as complementary or supplementary energy. This condition has begun to change, and in the future base-load steam generation will be required in increasing amounts.

SOUTHERN CALIFORNIA POWER SYSTEMS

Southern California is served by eight different power systems. The city of Los Angeles is supplied by its municipal Department of Water and Power, with energy generated (*a*) in its own hydro plants, (*b*) at Hoover Dam, and (*c*) in three steam stations. The cities of Pasadena, Glendale, and Burbank have municipally owned systems each supplied from its own steam station, in addition to purchasing hydro energy from Hoover Dam. The Imperial Irrigation District supplies energy generated by hydro along the All American Canal and by Diesel to the Imperial Valley area, and is now constructing a steam station. The metropolitan area surrounding San Diego is served by the San Diego Gas & Electric Company through a system which gets most of its energy from base-load steam, but obtains the remainder from wholesale purchases.

A portion of the eastern side of the coastal plain and some of the desert area is served by California Electric Power Company through a system supplied mainly by hydro plants in the Sierra Nevada Mountains, and by purchases from Hoover Dam and other agencies. The remainder of the Southern California area is served by Southern California Edison Company with energy supplied from Sierra Nevada and other hydro plants, from Hoover Dam, and from two large steam stations, a total system capacity of 1,583,000 kw. Further detail on these systems and their generating capacities are given in Table 1, and the map, Fig. 1.

POWER RESOURCES

At the end of the war the power resources of Southern California systems consisted of a total of 2,535,000 kw in operating capacity, installed by the eight agencies; 65.5 per cent of this capacity was hydro, 33.2 per cent was steam, and 1.3 per cent Diesel. Of the hydro, 39.6 per cent of the installed capacity was in 30 plants in the Sierra Nevada Mountains, or in plants utilizing water brought from those mountains; 59.4 per cent was at Hoover Dam, and 1 per cent was in miscellaneous smaller plants. Hydro energy output is ordinarily considered on three bases, namely, (1) that for a year of average water supply, (2) that for a year of low water supply, i.e., a dry year, and (3) that for a year of exceptional water supply, i.e., a wet year. These outputs vary from 6,579,000 kwhr for a dry year to 9,036,000 kwhr for an average year to 10,470,000 kwhr for a wet year. In other words, there is about 27 per cent less in a dry year than during an average year, and about 16 per cent more in a wet year than in an average. Except for the effect of the large storage at Hoover Dam, there would be 50 per cent less in a dry than in an average year.

This situation presents the basic need for steam generation up to the time the war began. Ordinarily, sufficient reserves are maintained either in stored water or steam-generating capacity to meet not only emergency system needs, resulting from breakdowns and seasonal and other peaks, but also for the full deficiency which occurs in a dry year. It is usually considered that about 40 per cent of a system's total capacity is required in steam generation to meet these needs.

WAR-END AND MORE RECENT CONDITIONS

During the war the presence of reserve capacity in Southern California led the War Production Board to locate several defense plants, including one each for magnesium and aluminum reduction, in the area. The effect of these additions was to reduce

---

[1] Manager of Engineering Department, Southern California Edison Company. Mem. ASME.

[2] "Postwar Planning for Steam Capacity in Northern California," by C. C. Whelchel and W. R. Johnson, published in this issue of the Transactions, pp. 237–246.

Contributed by the Power Division and presented at the Semi-Annual Meeting, San Francisco, Calif., June 27–30, 1949, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 49—SA-16.

the reserves to the point where load curtailment would have been necessary in a year of subnormal water supply. As the war ended, it was generally thought that postwar adjustment would reduce considerably the need for industrial power and hence allow a period during which to regain reserves by building new

TABLE 1 CAPACITY OF ELECTRIC GENERATING SYSTEMS SERVING SOUTHERN CALIFORNIA AS OF AUGUST 1, 1948[a]

	Total kva capacity of main units derived from name-plate rating			
	Hydro	Steam	Internal combustion	Total
California Electric Power Co...	55463	11250	293	67006
City of Burbank.............	...	25000	...	25000
City of Glendale.............	...	50000	...	50000
City of Los Angeles...........	157532	331430	...	488962
City of Pasadena.............	1500	62500	...	64000
Imperial Irrigation District....	18000	...	15000	33000
San Diego Gas and Electric Co.	...	253500	...	253500
Southern Calif. Edison Co......	508650	604981	37500	1151131
Hoover Dam[b]				
U. S. Department of Interior, Bureau of Reclamation......	1030000			1030000
Total	1771145	1338661	52793	3162599

---

[a] From figures compiled by California Public Utilities Commission.

[b] Of this capacity approximately 495,000 kva is operated for the Department of Water and Power, City of Los Angeles, and the cities of Pasadena, Glendale, and Burbank; 330,000 kva for Southern California Edison Company and 40,000 kva for the California Electric Power Company. In addition, 165,000 kva of capacity, which is installed for the primary use of the Metropolitan Water District of Southern California, is operated for the Southern California Edison Company and the California Electric Power Company during times that the full capacity is not required by the Metropolitan Water District.

TABLE 2 ADDITIONS IN CAPACITY TO ELECTRIC GENERATING SYSTEMS SERVING SOUTHERN CALIFORNIA, ARRANGED CHRONOLOGICALLY—1946 TO 1951

			Approximate name-plate rating of main generators, kva	
			Hydro	Steam
Southern California Edison Co..........	Long Beach	August, 1947		25000
City of Glendale......	...	October, 1947		25000
City of Los Angeles....	Harbor	November, 1947		81250
Southern California Edison Co..........	Redondo	March, 1948		70588
San Diego Gas & Electric.................	Silvergate	April, 1948		75000
Southern California Edison Co..........	Redondo	April, 1948		70588
Southern California Edison Co..........	Big Creek #3	April, 1948	35000	
City of Los Angeles....	Harbor	December, 1948		93750
City of Los Angeles....	Harbor	February, 1949		93750
City of Los Angeles....	Harbor	Spring, 1949		93750
Imperial Irrigation District................	El Centro	April, 1949		25000
City of Burbank.......	...	August, 1949		25000
City of Pasadena......	...	August, 1949		43750
Southern California Edison Co..........	Redondo	September, 1949		70588
Southern California Edison Co..........	Redondo	October, 1949		70588
Imperial Irrigation District................	Drop 4	March, 1950	12500	
San Diego Gas & Electric.................	Silvergate	October, 1950		75000
City of Los Angeles....	Owens Gorge	December, 1950	75000	
City of Los Angeles....	Owens Gorge	Spring, 1951	37500	
Southern California Edison Co..........	Big Creek #4	Late 1951	84000	
		Total.........	244000	938602

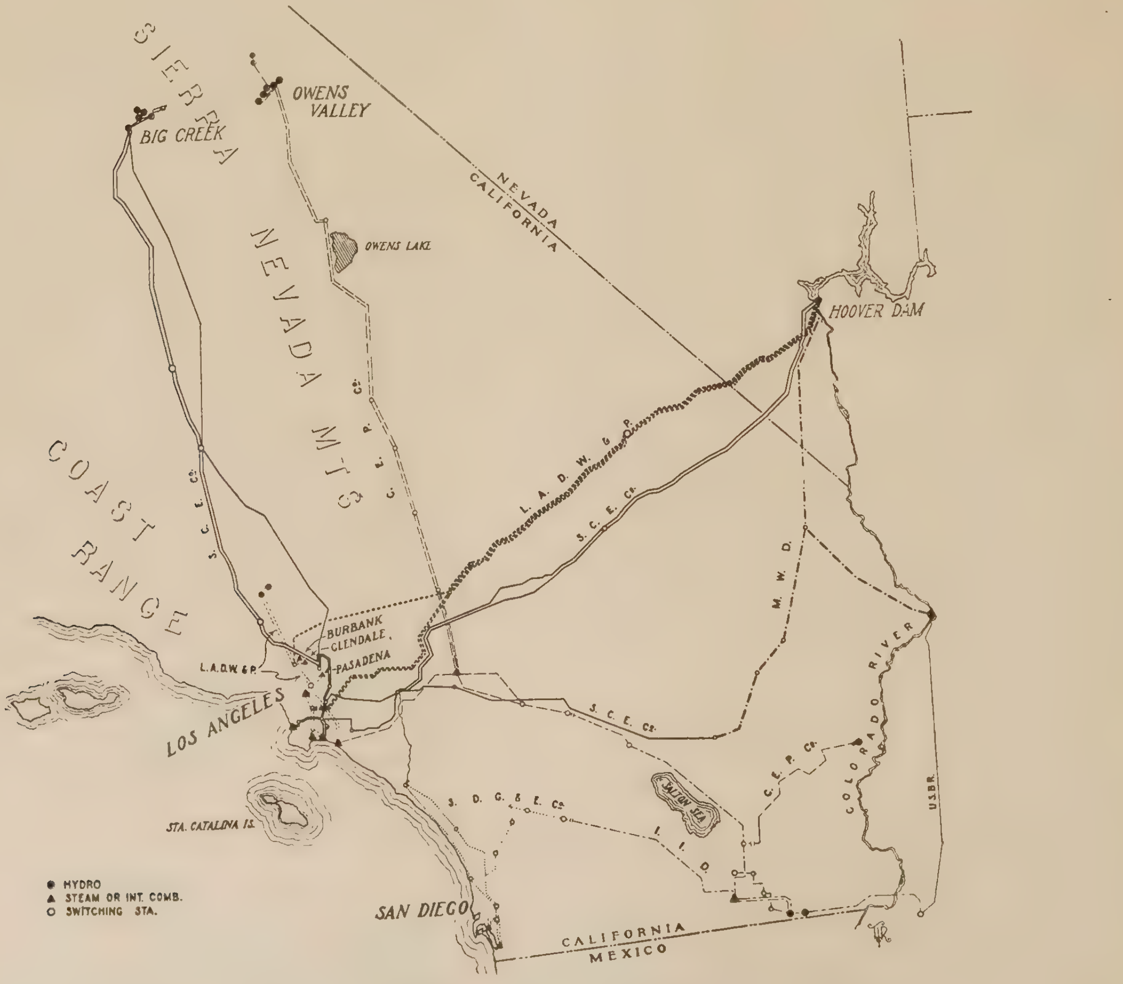


Fig. 1 General Scheme of Power Systems Serving Southern California

capacity. Although industrial load did drop for a time, its effect was more than offset by expanding energy use in other classifications.

By the first of 1946 it was apparent that in California, as well as elsewhere in the country, there would be no respite, but that the load, at least for some time, would increase faster than new generation could be built. From over the nation, orders were placed much faster than turbines, generators, and transformers could be built. Delivery promises increased from 16 months to 39. Strikes of all kinds during 1946 added months to delivery expectations.

Population Trend. In the meantime, population and loads were growing. Population in Southern California increased 47 per cent between 1940 and 1948. On the Southern California Edison Company's system during the 4 years between January, 1944, when voluntary wartime power rationing in this area ceased, and January, 1948, residential power use increased 52 per cent, commercial use increased 58 per cent, agricultural 75 per cent, and industrial 18.6 per cent. The over-all increase was 41 per cent.

Conditions in 1948. Combined with these influences, rainfall in the early part of 1948 was the lowest for many years, not only causing hydro production to fall off all over the state but also greatly increasing the demand for power for off-season irrigation pumping. In some areas, curtailment became necessary. Southern California had been successful enough in building new steam generation so that when that new capacity was combined with daylight saving and good rains, curtailment there was unnecessary. Also, by co-operative pooling of generating resources, it was possible to transmit large blocks of energy to the curtailed areas with the combined fortunate result that curtailment anywhere in the state was short-lived.

Southern California Conditions Affecting Steam-Power Expansion

Southern California steam-power expansion has been noteworthy for the following characteristics and developments:

1 The rapid increase in the steam-hydro ratio during the past 5 years.

2 The use of moderate steam pressures and temperatures.

3 Exclusive use of natural gas and oil but with space allowance for future coal.

4 The earliest use of outdoor plant construction.

5 The use of special operating methods, controls, and construction to obtain minimum-time load pickup from minimum to maximum.

6 Extensive use of high-pressure mechanical oil atomization.

7 Successful use of thermal control of marine fouling organisms.

8 The recent field rebuilding of three large-sized turbine generators during conversion from 50 to 60-cycle operation.

Hydro-Steam Ratio. As noted previously, at the end of the war the capacity serving Southern California was 2,535,000 kw consisting of 1,659,000 kw of hydro, 844,000 kw of steam, and a small amount of Diesel. Percentagewise for hydro and steam, this amounts to 65.5 and 33.2 per cent, respectively. Since that date (that is, to January 1, 1949) 34,000 kw of hydro and 388,000 kw of steam have been added, bringing the steam capacity currently to 41.7 per cent. Installations now in design and under construction will bring this ratio to 54.4 per cent hydro, 44.8 per cent steam by January, 1952 (Table 2).

Moderate Pressures Used. To the present time, two influences have tended to keep steam-power developments in Southern California in the moderate pressure and temperature range, namely, (1) relatively low cost of fuel because of the area being figuratively on top of large oil and gas fields, (2) availability of large quantities of relatively cheap hydro power and the resulting low load factor available for steam operation. Under the latter conditions, 35 per cent is a fair estimate of steam-power load factor. Accordingly, economy has required that new installations be designed and built for the pressures around 850 psi and temperatures of 900 F. With the increase in fuel-oil cost from 18 cents per million Btu to 36.1 cents between the war's end and January, 1949, and waning additional projects for hydro development, interest in more economical cycles is keener. One turbine is on order for 1250 psi 950 F, but no plant so far has been built in the area to utilize the reheat cycle.

Gas and Fuel Oil Only. Prior to the war, fuel gas was in abundant supply at costs of from 9 to 11 cents per million Btu, and oil at about 13 cents. With the increase in demand during the war for gas for industrial and domestic heating, except where gas could be obtained by direct purchases from the oil fields for boiler use, it became available only on a "dump" basis. This means that during warm weather when domestic use is light, and on week ends and holidays when industrial use is light, gas for power generation may be had in quantity, but there is no firm supply except where obtained by direct field purchase. Pipe-line importations are now being made from Texas, and others are in prospect, but again they afford no firm supply. As a result, power generation is principally supported by oil fuel. Some petroleum coke has been available but not in sufficient supply to be important.

As previously stated, no Southern California plant so far has been equipped with coal-handling and burning equipment although several have been built with space provision for bunkers, ash hoppers, pulverizers, and burning equipment.

A rough comparison of the cost of the three fuels under present (February, 1949) conditions is gas about 30 cents per million Btu, oil 31.2 cents to 32 cents per million Btu, depending upon the viscosity supplied, coal from Southern Utah about 44.7 cents per million Btu. With a large increase in fuel-oil stocks during recent months, oil prices for firm delivery have broken somewhat, backing off any immediate necessity for considering coal fuel seriously.

Early Use of Outdoor Construction. Mild weather and economy led to outdoor design for the two plants built by the cities of Burbank and Glendale about 10 years ago, two of the first to make use of such construction. The new Redondo Station of Southern California Edison Company has outdoor boilers, air preheaters, fans, and oil heaters. Three stations have been built on the unit principle with one boiler per turbine unit. Further use of this principle in the near future seems certain.

Quick Pickup Important. Because of the position in the load curve which steam power must take with respect to hydro, and hence of the frequent necessity of steam standing by at minimum loads for immediate operation in the event of an emergency, such as transmission-line or equipment loss, close attention is given in Southern California stations to the minimum load pickup time. The Redondo Station of Southern California Edison Company, for instance, is designed to increase from 5 per cent to full load in 15 sec. Other stations on systems depending upon long transmission lines for an important part of their resources have similar features.

High-Pressure Mechanical Atomization. With the increase in the use of cracking and other newer refining techniques, the production of fuel oil has passed from one in which this fuel was crude oil with the lighter and more valuable fractions removed by distillation to one where it is now a refinery residuum carrying tars, free carbon, and other heavy ends, plus considerable amounts of combustible and noncombustible residual chemicals. The resulting product while containing around 6,250,000 Btu per bbl possesses viscosities as high as 190 sec SSF and averages around 175 sec SSF for Bunker C fuel. This heavier fuel requires heating to between 80 F and 100 F to permit its conveyance from tanks to final heaters where temperature is increased to 200–210 F before supply to the burners. Further, this fuel produces deposits which adhere to closely spaced heat-transfer surfaces, particularly in superheaters, economizer and preheater sections which require washing down of external surfaces to permit maintaining anywhere near normal heat absorption.

Further, with the increased industrial use of fuel oil in the area, the need to minimize smoke emission, because of the special topographical and meteorological characteristics of the Southern California coastal area, has also made efficient combustion of first-order importance. To meet this need, a burner and supply system has been developed which utilizes a maximum burner supply pressure of 900 psi, obtained by screw pumps in the primary fuel-supply lines and booster pump connected to a recirculation line, which permits maintaining a burner pressure of not less than 300 psi at minimum load. The result is (1) wider range (1 to 20) between minimum and maximum boiler load, and (2) finer atomization, and hence better and more smoke-free combustion.

Redondo Station

Replaces Former Station. The Redondo Steam Station of the Southern California Edison Company is situated directly on the Pacific Ocean at Redondo Beach on an enlargement of the site of an earlier station which was considered uniquely efficient in its day. That original station was built a little more than 40 years ago and contained three 5000-kw General Electric generators, each driven at 100 rpm between two double angle-compound McIntosh & Seymour reciprocating engines supplied with steam at 175 psi and 475 F. The design and construction contract provided for a bonus payment for efficiency better than 170 kwhr per bbl of fuel oil (about 36,760 Btu per kwhr). Performance was 50 per cent above this requirement, earning a handsome bonus for its builders. Strangely enough, the advance in steam power generation was so rapid that within less than 4 years a turbine-driven generator was added, making the original units obsolete. Seventeen years later the station was sufficiently obsolete to put it only on the very peak of the load curve, and in another 9 years it was decommissioned and its boilers salvaged. The building was wrecked to permit construction of the new Redondo Steam Station.

Necessary to Utilize Previously Designed Equipment. Design of the new station was started early in 1946, at a time when (1) there was an unprecedented shortage of designers and draftsmen not only for station design but for equipment design and development, (2) strikes were tying up the entire production of the major electrical-equipment manufacturers and such basic industries as steel, coal, and transportation. Postwar adjustments were further delaying deliveries of power-plant equipment of all kinds.

In spite of these influences, it was necessary to have the new station in operation at the earliest practicable date. To offset these delays as much as possible, all major equipment was specified so that proposals could be offered for equipment which had been previously designed and preferably used, thus minimizing design and manufacturing time.

The joint AIEE-ASME Standards Committee had only recently issued its preferred standards for turbine generators, and some design work had been done on the largest or 60/66,000-kw size. Use of these standards and the estimated probable load factor for the station on the combined hydro system, fixed steam conditions at 850 psi and 900 F and turbine speed at 3600 rpm. One each General Electric and Westinghouse units were purchased at the outset, and one each additional late in 1946.

Boilers also were specified to fit radiant types which previously had been built and used successfully. Multiple and single boilers per unit were considered, but, although more expensive, multiple boilers were selected because, under system conditions, they afforded the greatest reliability and flexibility. To serve the first two units, four 400,000 lb per hr Babcock & Wilcox boilers were provided affording one spare. The final installation will include seven such boilers for four turbines, also affording one spare.

Station Arrangement. The station is laid out with the turbine generators arranged longitudinally along the center of a turbine building 73 ft wide and 485 ft long. A 45-ft-wide auxiliary and control bay parallels this room and forms the firing aisle for the seven boilers. One 200-ft reinforced-concrete stack is provided for each pair of boilers. The boilers, preheaters, forced- and induced-draft fans, secondary fuel-oil heaters, and deaerating heaters are outdoors on the inshore side of the turbine building. Because of the salt-spray-laden fogs which occur along the California coast, no consideration was given to outdoor turbine settings.

Cooling-Water System Design. The coast along which the Redondo Station is situated is exposed and subjected to heavy waves during the late fall, winter, and early spring. The original station had used an exposed vacuum-primed syphon pipe line with 9-ft drop legs set in the surf line to supply circulating water. This system was frequently in trouble from sand, seaweed accumulations, and storm damage. In designing the new station, every effort was made to afford a trouble-free circulating-water system. To obtain this objective, it was considered necessary (1) to keep the pipe lines below the ocean floor; (2) to keep the lines under positive pressure; (3) to avoid any exposed structure; (4) to extend the lines far enough seaward to set the intakes in water deep enough to afford 20 ft of water below low tide, and in order to avoid sand entrapment, to set the intake chime or inflow lip not less than 10 ft above the ocean floor; (5) to afford means of controlling marine fouling organisms which would make shutdown of the pipe lines for cleaning unnecessary. To effectuate the first conditions, both the intake and discharge lines are built of 10-ft-diam precast lock-joint concrete-pipe sections 16 ft in length laid below the ocean floor. The intakes are vertical tee sections 14 ft diam at the end of the 10-ft lines and set in 32 ft of water.

Marine Fouling Control. To control fouling organisms, an extensive investigation of chlorine injection was carried out, but previously it had been observed at the company's Long Beach Station that although the intake tunnels accumulated 4 to 6 in. of marine growth during the growing season, almost no growth occurred in the discharge tunnels. The only reason assignable for the difference was the difference in circulating-water temperatures. Using this long-time laboratory observation, it was decided to provide for fouling control both by means of chlorination and by periodically raising the temperature of the lines by alternately using each as a discharge. To check the possibility of satisfactory control by such means, the assistance of marine biochemists was sought, particularly that of Dr. D. L. Fox, of Scripps Institute of Oceanography.

The eventual solution was two separate 10-ft-diam lines extending an average of 1900 ft offshore where each is provided with an intake structure. These structures were set 200 ft apart, diagonal to the beach, to minimize recirculation. To permit alternate use as discharge and intake and likewise to enable chlorination without the need for chlorination lines extending seaward, a reversing chamber was provided on shore where flows may be reversed in 5 min and recirculated to build up any desired discharge temperature. By chlorinating the discharge line only and reversing pipe lines within the lethal chlorination cycle, all chlorine injection could be accomplished on shore.

Originally, it was considered that it might be necessary both to chlorinate and to elevate temperature and to put each line through such a cycle as frequently as each 8 hr. So far, control by heat alone has been so satisfactory that no chlorine has yet been used in the offshore lines. Furthermore, reversals only once each 2 weeks with temperature elevation to 95 F for 6 to 7 hr, to 102 F for $1^1/_2$ hr have killed off all organisms before they could mature to objectionable size. The arrangement of these features is shown diagrammatically in Fig. 2.

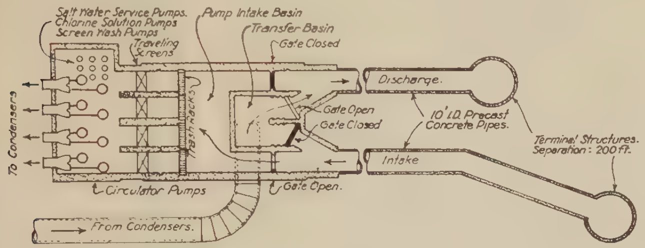


FIG. 2 SCHEMATIC DIAGRAM OF COOLING-WATER SYSTEMS AND PROVISIONS FOR REVERSING FLOW, REDONDO STEAM STATION

Cooling-Water Lines a Difficult Construction Problem. Construction of the cooling-water lines proved the most difficult part of the project but was successfully accomplished by Macco Corporation, the contractors, without delaying the operation of the station. Pipe laying was done entirely under water by divers served from a temporary trestle supporting excavating equipment, and a gantry crane for pipe handling. The courage and skill shown in handling the 42-ton sections suspended by slings and cable hoists in depths up to 50 ft, while subjected to fairly high seas, is a fine tribute to the construction fraternity, particularly since the visibility of a diver was often less than 1 ft, making work by touch only necessary most of the time.

Cooling-Water and Condensate System. Flow into and out of the reversing chamber is controlled by motor-operated welded steel gates, 7 ft × 10 ft. Automatic controls pair the gates during the reversing process. Trash racks with traveling rakes and stainless-steel revolving screens prevent the passage of fish or mats of seaweed into the pump chamber. Pumping to each condenser is by means of two vertical 25,000-gpm centrifugal pumps driven by two-speed motors for ease of adjustment from stand-by service to full load. Circulating-water temperatures vary from 55 to 70 F. The condensers are of Ingersoll-Rand manufacture and have 42,500 sq ft of surface. Water boxes and hot wells are divided. Condensate and condensate booster pumps are provided in duplicate with each able to return the condensate from 70,000 kw of load with all heaters in service. Three boiler feed pumps of 785 gpm capacity at 1100 psi are provided for each unit with a steam-driven pump for emergency use on either of each pair of units.

Boilers. All boilers are interconnected on the steam, the feedwater, and fuel sides. Each boiler is equipped with an outside heat-exchanger type of attemperator. Each also is provided with an extra large main steam drum (72 in.) to protect against carryover during water swell on quick load pickup. Each boiler is provided with automatic equipment which performs the following functions:

(*a*) Controls the flow of fuel to the burners.

(*b*) Controls the induced-draft-fan inlet louvers and outlet damper, except as modified by the fuel-air ratio controller to maintain a preset metered-fuel-air ratio.

(*c*) Controls the forced-draft-fan inlet vanes and outlet damper, except as modified by the furnace-draft controller to maintain a preset furnace draft.

Combustion Control. The combustion-control system was designed specifically for quick load pickup and also for wide load range. The following features were incorporated into the design:

1 Steam flow rather than steam pressure provides the impulse to increase the supply of fuel and air.

2 Air loading pressure is transmitted direct and simultaneously to the controls for fuel, air volume, and draft.

3 Control drives are of the "high-speed" type with extra large pilot valves and air-supply lines for rapid admission of operating air.

4 Combustion air is measured by a Venturi section in the air duct rather than being measured subsequent to combustion, thereby decreasing the lag in adjusting air flow to correspond to steam flow.

Fuel System. After primary heating, fuel oil is forced at about 725 psig by Sier-Bath screw pumps, to the main oil heaters adjacent to the boilers where it is raised to about 200 F, and forced to the "constant differential" pumps which raise the pressure to the burners to a full-load pressure of 900 psig and 200 F. The constant differential pumps afford a 20 to 1 load range. Automatic controls are provided in this system to start additional fuel-oil pumps in accordance with load demands, to control recirculation of oil to storage tanks, and to control flow of steam used for heating the oil. Furnace and burner protection is obtained through interlocking controls which shut off fuel to the burners if:

(*a*) Oil-supply pressure is low.
(*b*) Fan motors fail.
(*c*) Furnace has not been purged prior to "lighting off."

Turbine Generators. As previously stated, the turbine generators are the largest AIEE-ASME Preferred Standard, namely, 60,000/66,000 kw tandem-compound 850 psi, 900 F, 3600 rpm units, 13.8 kv voltage, 0.85 short-circuit ratio, built for either 0.5 or 15 psi hydrogen pressure. The heat balance for these units and the station is shown in Fig. 3. Each of the first two units has been tested and operated at loads to prove a safe output of 66,000 kw net.

Auxiliary Power. Two 6000-kw, 15-sec response, noncondensing house sets provide auxiliary power in the event a system voltage or frequency drop threatens the auxiliary power supply ordinarily supplied by two transformer banks connected to the system. On a drop in system voltage, these sets start automatically from standstill and come to full speed and voltage ready to carry full auxiliary load in 15 sec, but must be connected manually to the load at the time the auxiliary banks are disconnected from the system.

GENERAL

Each generator is connected to a 70,000-kva transformer bank. The output of the plant is controlled by a double bus switchyard and is transmitted over six 69-kv transmission lines to a major transmission substation about 5 miles away.

The station structures include an administration and service wing where all offices, shops, and chemical laboratory are housed. Because the region is subject to earthquakes, all structures are designed for a seismic coefficient of 0.2 gravity. This factor is twice that required for public school buildings in the state but is selected in the public interest to assure service continuity should a serious earthquake affect the area.

The station is equipped to burn either gas or fuel oil and space has been provided for pulverized-coal equipment should such become necessary. Some gas is available direct from the fields, but the principal fuel is oil. Storage is provided on the site for 220,000 bbl.

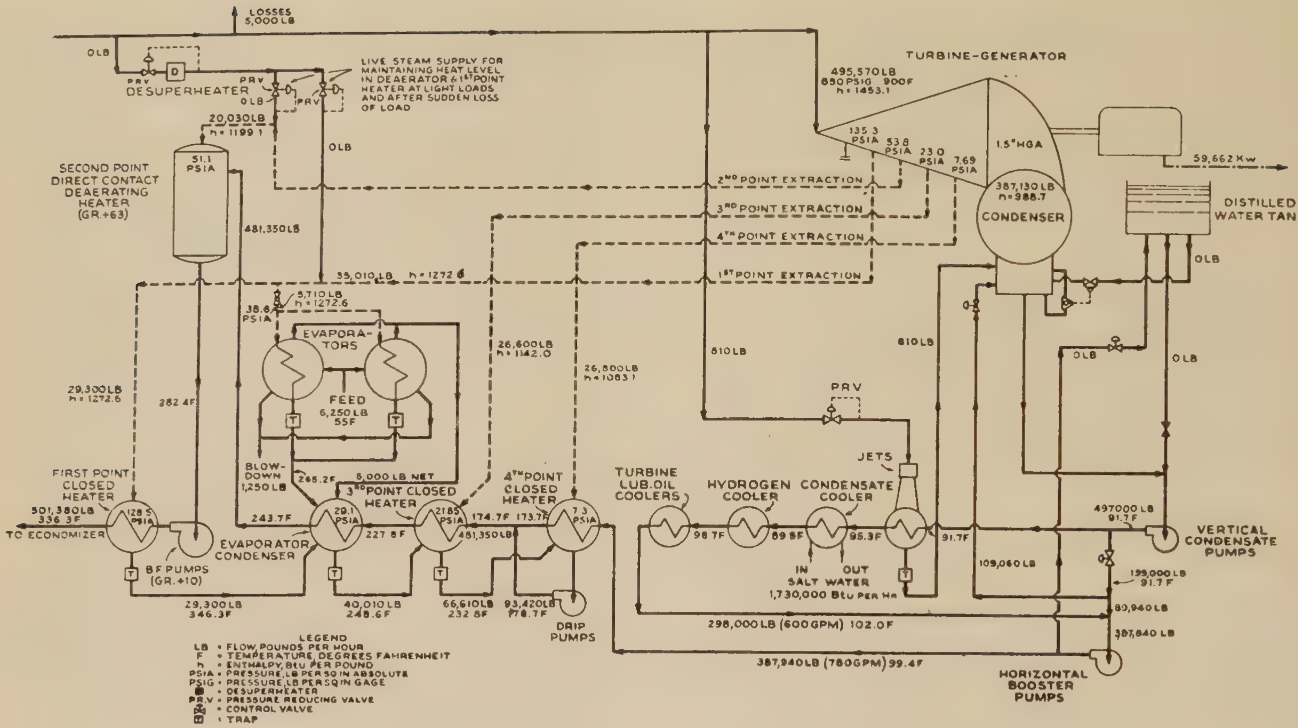


Fig. 3 Heat-Balance Diagram, Redondo Steam Station of the Southern California Edison Company

Because of the demand in California during late 1947, and early 1948, for every possible kilowatt of generating capacity, construction of the Redondo Station was under heavy pressure. The first turbine generator was erected from receipt of bedplate to first synchronization with the system in 70 elapsed days, and the second in 51 days. So urgent was the need for the station that it was put into operation as soon as the fundamentally essential manual controls could be completed, and several months ahead of the time when it would have been possible to provide a fully tested and automatically controlled plant. In spite of many threatened construction delays because of postponed equipment deliveries, the station was delivering energy to the system less than 18 months after excavation on the site was started. Operators, engineers, and construction men all co-operated to the latter end. Base-load operation has been the order of the day ever since the first unit was first put on the line.

The station was designed and constructed by the Stone and Webster Engineering Corporation in close collaboration with the Engineering and Operating Departments of the Southern California Edison Company.

Field Rebuilding of Large Units for Frequency Conversion

Incidental to conversion of most of the Southern California Edison Company power system for operation at 60 cycles instead of 50 cycles, it was necessary to rebuild two 100,000-kw 450-psi 750 F tandem-compound double-flow General Electric and one 60,000-kw 400-psi 700 F single-cylinder General Electric turbine generators in the field. In each instance it was necessary to replace the high-pressure casings, high and low-pressure spindle, front bearing pedestal, the governor, and the generator field. Further, to allow for the higher speed and greater steam space, it was necessary to reduce the stages in the larger machines from 21 to 17, and in the smaller machine from 20 to 17. This latter required machining new ledges for the new diaphragms.

All parts were manufactured in the factory and shipped to the field without benefit of check assembly of low-pressure parts. The machining of the intermediate and low-pressure casings was accomplished by field forces of the manufacturer, using a large boring bar operating in the bearings of the turbine. The first job was on one of the 100,000-kw machines and was completed during a $4^1/_2$ months' outage, the second was completed in 4 months, and the third in $3^1/_2$ months. Each machine went into service immediately after it was dried out and without clearance, balance, or other adjustment being necessary. All three were delicate and difficult shop and field manufacturing jobs most competently completed.

In the process of rebuilding, 10,000 kw additional capacity was built into each of the larger machines and nearly as much into the smaller one.

With an expansion of 760,000 kw in steam-generating capacity either constructed or in progress since the end of the war, the power systems of Southern California are determined that power supply to that area shall be in time and enough.

Discussion

F. X. Gilg.[3] The steam-generating equipment for Redondo as furnished by the writer's company consisted of four radiant boilers which were placed in service during the early part of 1948, and three additional radiant boilers of identical design which are about ready to be placed in operation. We would like to describe the principal features of the steam-generating equipment.

Fig. 4 of this discussion shows a cross section through these boilers, each of which is designed to generate 400,000 lb of steam per hr continuously at 875 psi and 900 F at the outlet of the superheater. The furnaces are of water-cooled construction, designed to burn either gas or oil. The contour of the water-cooled furnace makes it possible to adapt this furnace to pulverized-coal firing at some future date, if this becomes desirable.

---

[3] Application Engineer, The Babcock & Wilcox Company, New York, N. Y. Mem. ASME.

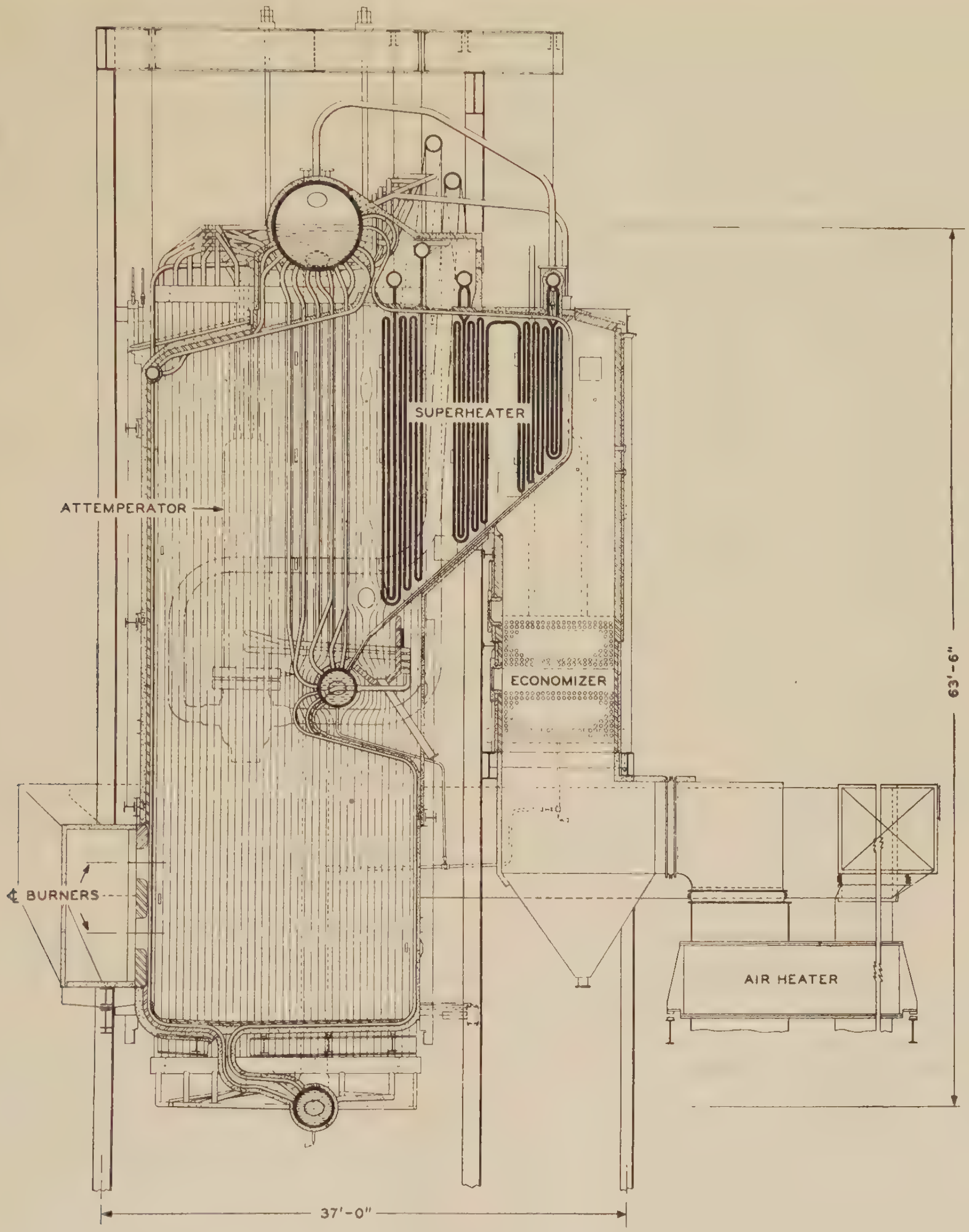


Fig. 4 Cross Section Through Redondo Station Steam Generator

The boiler and superheater tubes are arranged on wide centers to minimize tube fouling from oil-fuel slag. The first rows of boiler tubes are 4 in. OD on 18 in. centers, while the remaining three rows are 4 in. OD on 9-in. centers. The first bank of superheater tubes consists of $2^1/_2$- and $2^1/_4$-in. tubes on 7-in. centers. The second and third banks of superheater tubes are $2^1/_2$ in. OD on $3^1/_2$-in. centers.

The steam drum is 72 in. diam to provide water storage and space for water-level rise during quick load pickups. The steam drum contains cyclone separators and steam scrubbers so that the solids content of the steam leaving the boiler will not exceed 1 ppm even during sudden load pickup.

Steam temperature is maintained at 900 F from 250,000 to 400,000 lb of steam per hr by means of a shell-type attemperator connected between the first and second sections of the superheater through which a portion of the superheated steam will flow, giving up some of its heat to boiler water which circulates around the outside of the tube bundle in the attemperator. The amount of steam that goes through the attemperator tube bundle is controlled automatically by a rotary valve. Quick response of temperature control is an important factor in steam-generating units designed for quick pickup of load.

The economizer is of the continuous-tube type, discharging into the steam drum through cyclone separators.

To facilitate the cleaning of troublesome deposits from superheater tubes, the floor under the superheater is arranged with a drain hopper so that the superheater surfaces can be washed with water during out-of-service periods. The economizer and air heater are also arranged so that they can be washed during out-of-service periods.

The expected efficiency of 81.41 per cent on gas fuel and 85.46 per cent on oil fuel have been exceeded on heat-balance tests which showed 82.24 per cent efficiency on gas, and 86.42 per cent efficiency on oil.

C. L. Hathaway.[4] Mr. Chadwick has presented a very clear picture of the conditions in Southern California relative to the electrical demand in this area and to the facilities for meeting this demand. He has also brought to our attention many very interesting design details of the new Redondo Steam Plant. Of much interest is the method of controlling the growth of marine life in the circulating-water system, a very serious problem which the pioneering efforts of his company have solved successfully.

This discussion of Mr. Chadwick's paper will be limited to the following comments concerning the most southerly coastal portion of the area considered, that of San Diego. Several years before the war it became evident that the power facilities of the San Diego Gas & Electric Company would have to be expanded to meet the rising demand. This was due to the very great increase in the population of the area with its increase in domestic load requirements, to the rapid development and growth of the aircraft industry, and to the expansion of the needs of the United States Government. The company had at that time only one generating plant, Station B. This is a low-pressure steam station operating at 235 psi 700 F, located in downtown San Diego. Expansion of Station B was not advisable. The cost of increasing the circulating-water supply would have been prohibitive, and property values were comparatively high. A topping installation was considered, but the idea was abandoned because the load cycle did not lend itself well to this type of operation and because building alterations would have been excessive. In addition, any appreciable increase in the station's generation would have required extensive changes in the electric-transmission system in this area. For these and other reasons, it was deemed advisable to build a new station.

The site of the Silver Gate Station was chosen for several reasons; property values were reasonable, the station could be built near the San Diego Bay so that circulating-water problems would not be serious, and the location would be in a central part of the distribution system, with close proximity to the rapidly growing South Bay area.

Work was begun in 1941, and the station was placed in commercial operation in March, 1943. This plant consists of a 35,000-kw General Electric turbine, driving a 50,000-kva hydrogen-cooled General Electric generator at 3600 rpm. Steam is supplied at 850 psi and 900 F by two 200,000-lb per hr Babcock & Wilcox type F integral furnace boilers.

Before unit No. 1 was placed in service it was apparent that it would not be sufficient to meet the rapidly increasing load. Therefore, in December, 1941, the company placed an order for the second unit (50,000 kw). This was later canceled by government order so that all manufacturing facilities might be directed exclusively to further the war effort. It was not until late in 1945 that it was possible again to place orders. Work of installation was begun in 1946, and unit No. 2 was placed in service in April, 1948.

The delay in the installation of unit No. 2 due to the war, and the unexpected increase in load at the cessation of war activities has made unit No. 3 an immediate necessity. Plant construction is now under way, and it is hoped that it may be placed in service early in the summer of 1950. This unit will operate at 1250 psi 950 F. This increase was deemed advisable in view of the increasing cost of fuel. Studies are now under way for unit No. 4, which will be the last at this site. Ultimate capacity of the station, based upon turbine ratings, will be 185,000 kw.

Fig. 5 herewith is an average of the actual week-day load conditions of the San Diego Gas & Electric Company for the month of January, 1949. This shows the peculiarities of loading in a comparatively small system in which the load is largely domestic. This load demand must be met by an all-steam system, as the company has no hydroelectric generating stations. Serious operating problems are introduced by the extremely high rate of load change, particularly when going from the minimum to the morning peak. This represents an increase of 150 per cent in $2^{1}/_{2}$ hr. This figure is not impressive if we consider the expected performance of a true stand-by plant, but it must be remembered that this is a daily occurrence and that a large part of this system pickup must be handled by the 99,000-kw low-pressure station, whose increase is in the nature of 500 per cent.

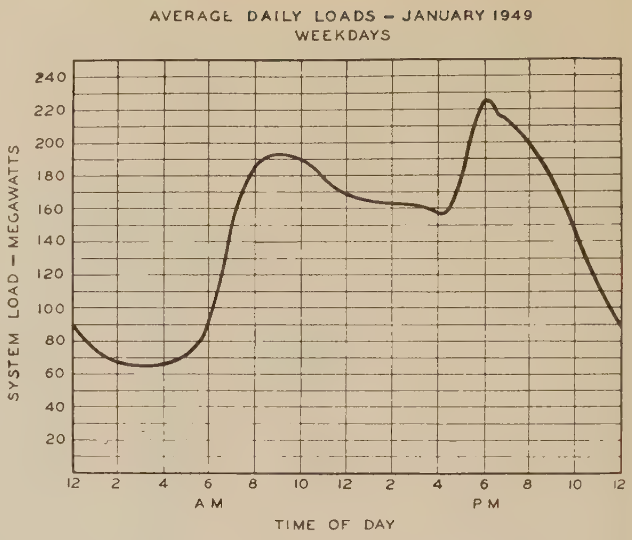


Fig. 5 Average Daily Loads on Weekdays, January, 1949

An explanation of the San Diego Gas & Electric Company's recent development in steam generation can best be made by describing Silver Gate unit No. 2. In listing the various items of equipment, comment will be made upon certain features which have been changed from the design of unit No. 1 and certain other points of interest.

The turbine is a General Electric 3600-rpm "Handbook" tandem-compound machine with double flow in the low-pressure section. It is rated at 50,000 kw with an overload rating of 25 per cent, or 62,500 kw. It was designed for and uses four points of extraction for feedwater heating. The hydrogen-cooled General Electric generator is rated at 60,000 kw at 80 per cent power factor (PF), or 75,000 kva. This rating is for $^{1}/_{2}$ psi hydrogen pressure. The main exciter is driven through a reduction gear coupled to the generator shaft and runs at 1775 rpm. It is a 220-kw 250-volt unit. The pilot exciter, a 4-kw 250-volt unit, is overhung from the end of the main exciter shaft.

Fig. 6 of this discussion is a heat balance for unit No. 2. The values shown are the turbine-generator manufacturer's estimates for a generator load of 62,095 kw (approximately $^{5}/_{4}$ turbine rating). This is a simple system with four-point regenerative feedwater heating. Generator hydrogen and exciter air are cooled by the full condensate flow from the hot-well pump. Condensate flow through these coolers and through the air-ejector condensers is increased during start-up, and at low loads by recirculation to the condenser. Make-up to the system is supplied by an evaporator receiving steam from the second turbine extraction point (10th stage) and discharging its vapor to the deaerating heater. The deaerator pressure is not held constant, but swings with that of the turbine third extraction point (14th stage) as load on the unit changes.

The turbine lubricating-oil system of unit No. 2 differs from earlier units in that there is no shaft-driven pump. Instead, there

---

[4] Superintendent, Electrical Production, San Diego Gas and Electric Company, San Diego, Calif.

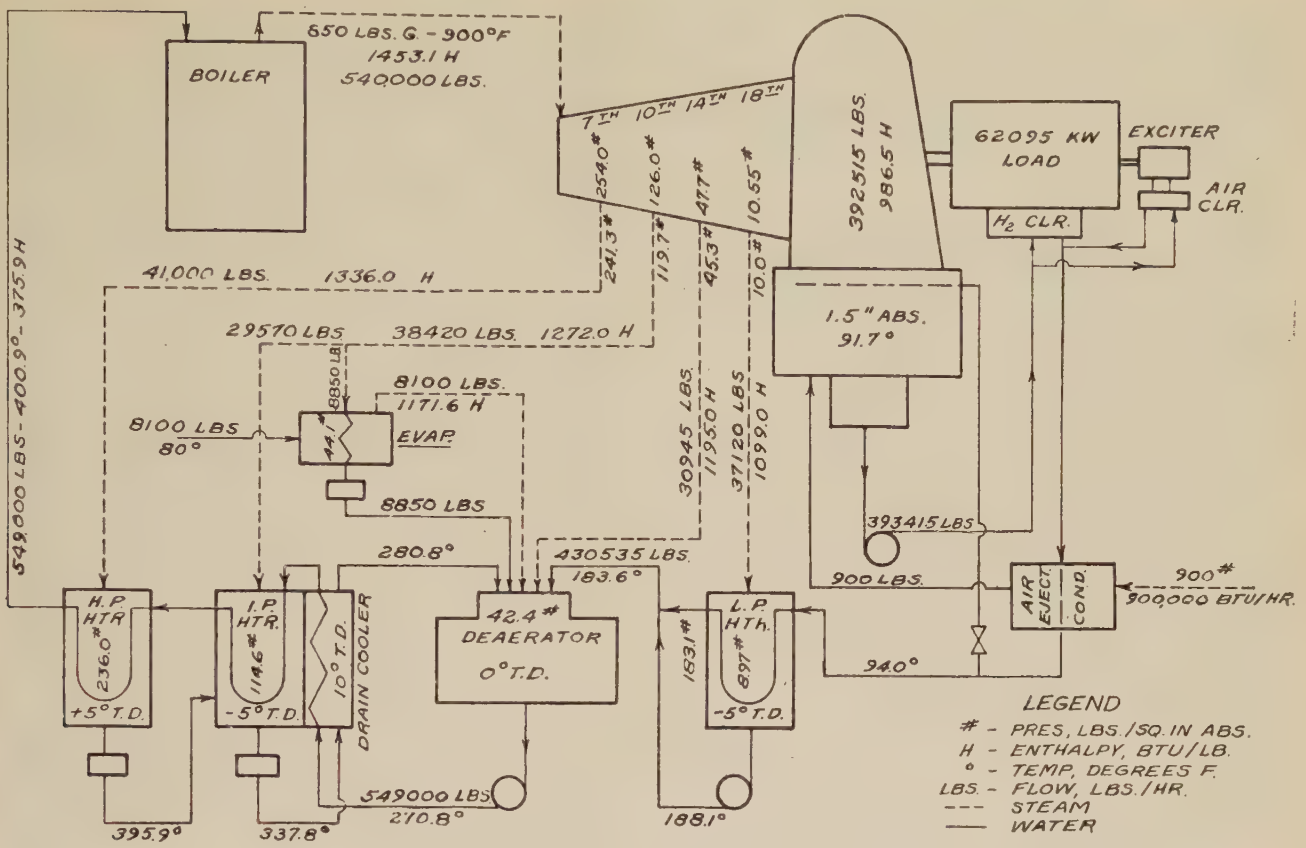


Fig. 6 Heat-Balance Diagram Silver Gate Unit No. 2
(60,000 kw, 3600 rpm, 0.8 pf, hydrogen-cooled generator, 850 psig, 900 F, $1\frac{1}{2}$ in. Hg abs.)

are two motor-driven pumps, so interlocked that the reserve pump will start on an outage of the preferred pump. There is also a steam-driven pump, which starts on low pressure, for emergency use. This pump rotates continuously at low speed, as a small amount of steam is fed through its turbine to keep it warm and available for immediate service.

The boilers of unit No. 2 are two Babcock & Wilcox integral furnace-type F which supply steam at 850 psi 900 F to the turbine. Each has a capacity of 300,000 lb per hr with an overload capacity of 10 per cent for 4 hr. As with unit No. 1, each boiler has two superheaters, with an externally mounted attemperator in series between them. Considerable difficulty has been experienced with slag formation on the tubes of the unit No. 1 primary superheaters when oil fuel has been used. These deposits have been so heavy that actual bridging between tubes has resulted. This has caused excessive draft loss, requiring reduction in boiler load. The material on the tubes is hard and tough so that an extended outage is necessary to remove it mechanically. Use of natural gas as fuel tends to remove the deposit, but as gas is available only in certain seasons and then in limited supply, it cannot be depended upon to maintain cleanliness. Although the deposit is softened by water and is readily washed off, it has not been deemed advisable to water-wash these superheaters for fear of damage to the refractories and possible corrosion of the superheater headers and the boiler mud drum. For this reason the design has been changed for unit No. 2. The superheaters are of the pendant type with the headers above, and hoppers have been provided to drain off the wash water properly. It is felt that these superheaters can be quickly and effectively cleaned. In addition to this, the tube spacing in the primary superheater has been increased to 6 in. between centers, giving much greater space for gas passage than in the older superheaters where tubes were spaced at 3 in. This has necessitated locating part of the primary superheater in the second gas pass of the boiler.

The waterwalls of the furnaces of the unit No. 2 boilers are the touching-tube type, instead of the stud-tube design of unit No. 1. This greatly reduces the amount of refractory required and increases materially the radiant-heat absorption. Boiler convection surface is reduced somewhat by the increased size of the superheater.

Boilers of both unit No. 1 and unit No. 2 are equipped with tubular air preheaters. The preheaters of unit No. 1 have horizontal tubes with the air on the inside and the gas on the outside. During the war period it was necessary to burn fuel oil with a comparatively high sulphur content. The resulting high-dew-point flue gas caused sufficient condensation on the cold-end tube sheet for chemical attack, so that a large number of the tubes were cut completely off at this point. In order to avoid this trouble in unit No. 2 preheater, the design was changed materially. The hot gases are on the inside of the tubes with the air outside. The preheater is divided into two sections, a main section and a short-tube cold-end section. It was thought that if high-dew-point conditions were again encountered it would be a simple matter to replace tubes in the cold-end section or, if necessary, alloy tubes might be used. As a further protection a cold-air by-pass was provided to control the cold-end temperatures at low load operation.

Both the forced-draft and induced-draft fans of unit No. 1 use vane control. Considerable trouble was experienced in the vane bearings of the induced-draft fans. Because of the high temperatures to which they were exposed the lubricant carbonized and froze the balls in position, which resulted in operating difficulties and in high maintenance. Although this condition has been largely removed by the use of silicone lubricants, it was decided to use a different control for the induced-draft fans of unit No. 2. These fans are driven at variable speed through hydraulic couplings. Additional control is obtained by the use of dampers in the fan ducts.

Fuels used at Silver Gate are natural gas and industrial fuel oil Bunker D. In unit No. 1 this oil is delivered to the control valves at 280 psi. A centrifugal pump which maintains constant differential of 100 psi between the burner supply and return, raises this to about 350 psi at maximum demand. For unit No. 2, centrifugal booster pumps raise the pressure to 660 psi at the control valves, and a 250 psi differential is maintained at the burners, giving a full-load maximum of about 900 psi. This increase in fuel-oil pressure is to permit the use of smaller burner tips and thus to obtain better atomization and better combustion at low loads. It is planned to convert unit No. 1 to the same pressure levels, and a comparable change will be made at Station B.

Fuel-oil storage is by three steel-lined concrete tanks located beneath the 12.5-kv substation structure. Each tank has a capacity of approximately 5300 bbl, with a normal net of a little over 4000 bbl. Delivery to these tanks is by pipe line from the oil company's tanks or direct from tankers. The storage afforded by these tanks is not deemed adequate, so it is planned to provide an 80,000-bbl tank for additional reserve.

High-pressure steam piping in unit No. 2 uses welded joints throughout. All joints were checked by gamma ray. This is a change from unit No. 1 piping where octagonal ring-flanged joints are used.

Bearing and jacket cooling at Silver Gate is supplied by fresh water. This is a closed system, common to both units, in which city water is circulated through salt-water heat exchangers. The fresh water is treated to maintain a noncorrosive condition. Make-up to this system is by condensate drains, so that the solids concentration is less than that of the city water. The only use of salt water is in these exchangers and in the main condensers.

Fig. 7 herewith is a diagram showing the station's hookup with the distribution system. Electric generation is at 12,500 volts. Each generator feeds a ring bus which is connected on one side to the two main buses of a 12.5-kv substation, and on the other to the transformer banks for the 69-kv system. Station auxiliaries for each unit are supplied at 2400 volts by two transformers, one from the generator bus and one from a feeder from the 12.5-kv substation. The larger units operate at 2400 volts, the smaller ones at 220 volts.

The Silver Gate Station is not of the outdoor or semioutdoor type, in that both the turbine and boiler rooms are enclosed. The partially open type of plant was considered, but the advantages of the closed building were believed to more than offset its increased cost. It might be well to mention a few of the points considered in making this decision. A slight increase in operating efficiency results from the partial recovery of radiation heat losses. This is effected by placing the forced-draft fans on the top floor of the boiler room. Next, it is possible to do maintenance work in any part of the plant on short notice, regardless of weather conditions. This is of considerable importance in a small all-steam system, such as that of the San Diego Gas & Electric Company. Further, morale of the operating personnel is definitely better with the closed plant. This is a point that is always considered but usually not credited with sufficient importance. There are other advantages to the enclosed station; such as, greater ease in maintaining plant cleanliness, less chance of neglect of routine operating duties, etc.

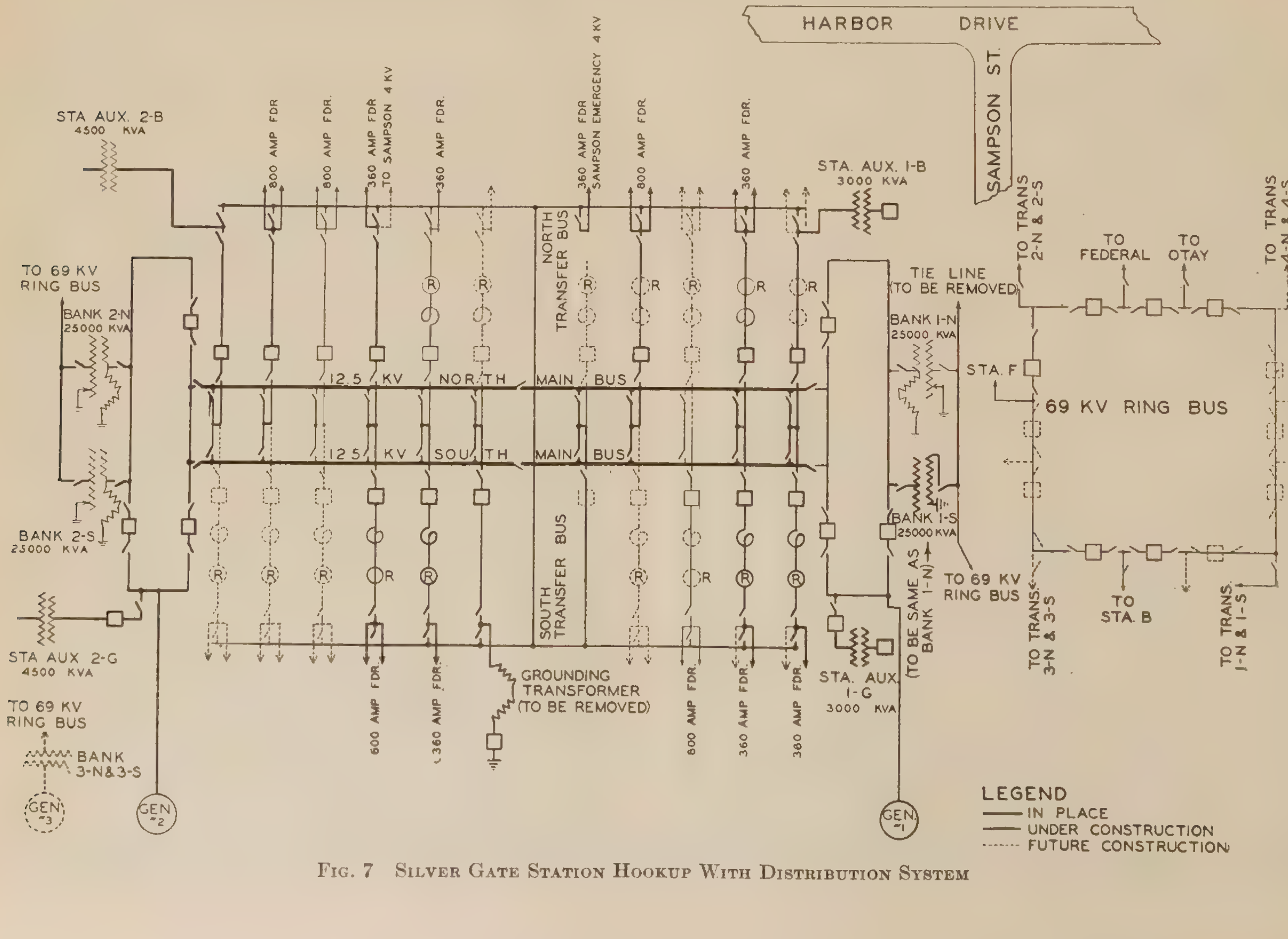


Fig. 7 Silver Gate Station Hookup With Distribution System

TABLE 3 PRINCIPAL POWER-PLANT EQUIPMENT AT SILVER GATE STATION UNIT NO. 2

TURBINE GENERATOR UNIT	
Turbine 50,000 kw, 3600 rpm, tandem-compound, double-flow, condensing steam turbine 850 psi, 900 F, $1^1/_2$-in. exhaust; 4 bleed points.	General Electric Company
Generator 60,000 kw, 0.8 pf, 12,500 volt, 3 phase 60 cycle, hydrogen cooled. Main exciter direct connected through reduction gear, 1775 rpm, 220 kw, 250 volt, pilot exciter 4 kw, 250 volts.	General Electric Company
Condenser Two-pass, divided water box, surface type with 40,700 sq ft effective surface. Tubes are of arsenical aluminum brass, tube sheets of rolled muntz metal.	Worthington Pump & Machinery Corporation
Circulating-water pump Two 36-in. vertical, 21,000-gpm, 390-rpm, motor drive, 150 hp, 2300 volts.	Worthington Pump & Machinery Corporation
STEAM-GENERATING EQUIPMENT	
Boilers Two integral-furnace type, 875 psi 900 F 300,000 lb per hr; overload capacity 330,000 for 4 hr. Heating surface, boiler and furnace: 21,500 sq ft superheater, pendant type, three steam passes. Temperature control by external attemperator.	Babcock & Wilcox Company
Air preheater Vertical tubular, 2 sections, 2 gas passes, gas in tubes, 3 air passes. Heating surface 44,674 sq ft.	Babcock & Wilcox Company
Forced-draft fans One per boiler, 101,000 cfm at 120 F, 1180 rpm, variable vane control.	Buffalo Forge Company
Induced-draft fans One per boiler, 181,000 cfm at 435 F. Control is by variable-speed drive through hydraulic coupling, and dampers.	Buffalo Forge Company
Fuel burners Four per boiler, type H-26, combination gas and oil. Oil burners are wide-range mechanical-atomizing with constant differential of 250 psi and a maximum top pressure of 900 psi.	Peabody Engineering Company
Safety valves Three per boiler, 4 in.	Crosby Steam Gage & Valve Co.
Combustion control system Fuel flow-air flow on both fuels; temperature control.	Bailey Meter Company
CONDENSATE AND FEEDWATER SYSTEM	
Condensate pumps Two 3-stage 1183 rpm, 963 gpm	Worthington Pump & Machinery Corporation
Feedwater heater No. 1 Low-pressure, vertical, 4-pass, 1915 sq ft surface. Steam from 18th stage extraction.	Foster Wheeler Corporation
Feedwater heater No. 2 Deaerator, 750,000 lb per hr, storage capacity 124,000 lb. Steam from 14th stage extraction, evaporator vapor, etc.	Cochrane Corporation
Feedwater heater No. 3 High-pressure, vertical, 4-pass, 2506 sq ft surface; has drain cooler section. Steam from 10th stage extraction.	Foster Wheeler Corporation
Feedwater heater No. 4 High-pressure, vertical, 4-pass, 2206 sq ft surface. Steam from 7th stage extraction.	Foster Wheeler Corporation
Evaporator Type H.P., single effect, with preheating and degassifying section. Capacity 12,000 lb per hr. Pressures: coils, 250 psi; shell 50 psi.	Foster Wheeler Corporation
Boiler feed pumps Three 850 gpm, 3568 rpm, 2880 ft total head, motor-driven, barrel-type casing.	Ingersoll-Rand Company
Feedwater regulators Three-element control.	Bailey Meter Company
PIPING, VALVES, AND FITTINGS	
Piping contractor	Midwest Piping & Supply Co.
Main steam line	Chrom-moly ASTM A-280-46T
Main steam valves	Crane Company
Stop-check valves	Edward Valve Company
Blow-off valves	Edward Valve Company
Boiler feed pump check valve	Chapman Valve & Mfg. Co.
Small high-pressure valves	Edward Valve Company
Steam traps	Henry Vogt Machine Company Armstrong Machine Works Yarnall-Waring Company
ELECTRICAL EQUIPMENT	
Main transformers Two TCUL, 12.5/69 kv 25,000 kva	Westinghouse Electric Corp.
Station transformers Two 12.5/2.3 kv, 4500 kva	Westinghouse Electric Corp.
Station auxiliary transformers Two 2400/240 volt, 750 kva	Pennsylvania Transformer Co.
Station lighting transformers Two 2400/240/120 volt, 200 kva	Kuhlman Electric Company
Generator oil circuit breakers Two 15 kv, 3000 amp	Kelman Electric & Manufacturing Company
2400-volt switchgear Indoor type, metal clad	General Electric Company
220-volt power centers	ITE Circuit Breaker Company
Motors:	
Boiler feed pump	Westinghouse Electric Corp.
Fans	General Electric Company
Circulating-water pumps	Westinghouse Electric Corp.
Hot-well pumps	Westinghouse Electric Corp.
Compressors	General Electric Company
MISCELLANEOUS EQUIPMENT	
C.W. traveling screens	Chain Belt Company
Control air compressors	Ingersoll-Rand Company
Heater drain pumps	Ingersoll-Rand Company
Fuel-oil booster pumps	Byron Jackson Company
Sump pumps	Yeomans Brothers Company
Differential oil pumps	United Iron Works

The Silver Gate plant is a windowless building except for the office bay. Air is supplied by a ventilation system which takes in outside air at the top-floor level. This gives a supply of fresh air to the station more nearly free of dust than would be provided by windows or louvers at near street elevation.

As a one-unit plant, the Silver Gate Station was operated by a four-man crew, in addition to a part-time electrician. This consisted of one engineer, one fireman, and two auxiliary engineers. When unit No. 2 was placed in service an additional fireman was provided, bringing the total to five men.

The engineer, in addition to his duties as turbine operator, handles all the electrical operations frequently performed in a separate control room. Synchronizing and breaker operation are controlled at the turbine-generator panel. This of course does not include operation of the substation which is by supervisory control by the system load dispatchers.

Table 3 gives a list of the major equipment of Silver Gate Station unit No. 2.

B. G. HUME.[5] The City of Los Angeles' Harbor Steam Plant is located adjacent to the Los Angeles Harbor. This plant was designed, built, and is operated by the Department of Water and Power, a municipally owned corporation serving the 450 square-mile area within the limits of the City of Los Angeles. This location was selected for a plant site as it afforded an ample supply of cooling water for a plant of large capacity, combined with proximity to the center of the city's industrial load.

Studies made in 1939, of the anticipated growth of load on the system, indicated that it would be necessary to add additional steam-generating capacity to the system by about 1943. This study of system requirements indicated that more stand-by steam power would be needed in the form of spinning reserve, and that this type of operation probably would change to base-load operation as the number of steam units on the system increased. Therefore it was decided to build a new steam plant, the first two units of which would provide spinning reserve and, when required, carry system peaks or base load.

Accordingly, orders were placed for two 81,250-kva units, the first to go into operation in 1943, and the second about a year later. The war interfered with this program, however, and the second unit did not go into operation until late in 1947.

By 1946 it became evident that unless considerable additional generating capacity was added as soon as possible, there would be a power shortage in Los Angeles. Orders were then placed for three additional units having a capacity of 93,750 kva each.

The curves in Fig. 8 of this discussion show the growth of load in terms of kilowatthours generated and kilowatt peak, respectively. These curves are extended beyond the year 1948, showing the probable growth of the load on the Department's system to and including the year 1952. Unit No. 2 was completed in November, 1947, just in time to prevent a serious power shortage on the system. Likewise, units Nos. 4 and 3 were completed in December, 1948, and February, 1949, respectively, in time to enable the department to help alleviate the shortage of power in California.

There are five turbine-generator units now installed in this plant. The fifth unit will go into service in the near future. This will complete the plant to its ultimate rated capacity of 355,000 kw on the main units. Unit No. 1 and unit No. 2 each consists of a 65,000-kw, 0.8-power factor (pf), 3600-rpm tandem-compound

---

[5] City of Los Angeles, Department of Water and Power, Los Angeles, Calif.

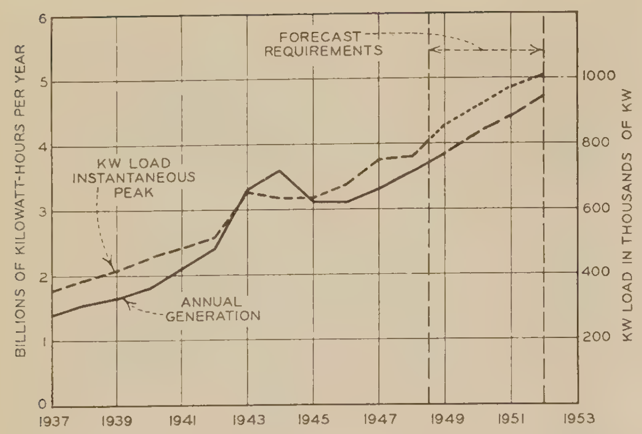


Fig. 8 Power System of the City of Los Angeles Department of Water and Power

turbine connected to a 81,250-kva, 60-cycle, 3-phase, 13,800-volt hydrogen-cooled generator. Units Nos. 3, 4, and 5, each consists of a 75,000-kw, 0.8-pf, 1800-rpm, single-cylinder turbine connected to a 93,750-kva, 60-cycle, 3-phase, 13,800-volt hydrogen-cooled generator. The nominal steam pressure and temperature at rated capacity is 850 psig and 900 F. Each turbine exhausts into a twin condenser, consisting of an exhaust trunk connected to the turbine exhaust which divides the steam and discharges it into two one-half size condensers located on either side of the foundation. This design permits low center of gravity and low siphon leg on the cooling-water outlet.

Circulating water is supplied to each twin condenser unit by two vertical mixed-flow pumps driven by vertical splash-proof motors.

The plant requires approximately 300,000 gpm cooling water. This water is taken from slip No. 5 of the harbor by means of two 8-ft-diam precast concrete pipe tunnels 1200 ft long and discharged into the West Basin through two similar tunnels approximately 1400 ft long. The temperature of the water varies from 55 F in the winter to 70 F for a short period of time in the summer.

Each turbine is supplied with steam from one steam-generating unit. The first two boiler units installed have a rated capacity of 570,000 lb of steam per hr and a 2-hr overload capacity of 760,000 lb per hr. The remaining three steam-generating units have a continuous rating of 650,000 lb of steam per hr, with a 2-hr overload rating of 825,000 lb per hr. These units are equipped to burn oil and gas and can be converted to burn pulverized coal at reduced rating if it should be necessary to burn the latter fuel at some future time.

All auxiliaries are motor-driven except for the emergency boiler feed pumps which are driven by steam turbines. The cycle is a straight regenerative cycle with four extraction points.

Horizontal feedwater heaters are arranged vertically, one above the other in the auxiliary bay, to provide positive drainage at low loads and to take care of surges when the load is suddenly increased.

As the first two units were to be designed for either base-load, peaking, or stand-by service, they would have to be capable of carrying the least possible load when there was plenty of hydroelectric power available and to pick up full maximum load instantly at the time of an outage and carry it until such time as the system might be restored to normal, or to carry a base load for protracted periods of time during dry years or whenever the system load exceeded the capacity of the hydroelectric plants. For the latter reason, these two units were designed for good economy when operating on base load. The wisdom of building economy into this type of plant has been proved during the past power shortage, as it was necessary to operate most of the time on a base load. The performance of 10,900 Btu per net kwhr at the most efficient load has been realized from these units.

When it became necessary to add the last three units, it was decided to eliminate the extreme requirements for stand-by operation, imposed on the first two units, by eliminating the excess fan and fuel-burning capacity built into the first two units. While the last three units afford considerable stand-by power, if required, they have not been designed for the very low load operation. This was considered unnecessary because of the greater proportion of steam to hydroelectric power existing at the later date.

Units Nos. 1 and 2 have been designed to operate on a minimum block load of 5000 kw indefinitely and, when the occasion demands, assume full rated load instantly. In order to accomplish this result, it was necessary to provide the following features in the design of the plant:

1 Absolute minimum carry-over in the steam under all conditions of operation.
2 Wide-range burners.
3 Positive, fast-acting fan control over a wide range.
4 Positive superheat control over a wide range.
5 Positive, fast-acting automatic combustion control
6 Positive heater drainage at all loads.
7 Automatic boiler feed pump control.

The boilers were designed for a minimum water-storage capacity of 3500 cu ft. The controls were designed to maintain a constant steam pressure in the steam drum. A dry drum was provided to prevent carrying over of any quantity of water to the turbine during periods of quick pickup and also to prevent carry-over of solids into the superheater and turbine during normal operation. This design provided maximum heat storage in the boiler and maximum steam pressure at the throttle when the unit was operating on stand-by load.

The fans and burners were designed to have about one and one-half times the required full-load capacity in order that the unit could be fired at a rate high enough to enable the boiler to supply steam for the highest possible quick-pickup load and at the same time recover full steam pressure in the least possible time.

These requirements made it necessary to install oil and gas burners having a maximum range of 15 to 1, and fan drives that could control the speed of the fans accurately enough for the automatic combustion control to function properly when the unit is carrying a 5000-kw load. At the same time, these fan drives had to be able to increase the rate of firing the boiler fast enough to enable the turbine to assume full rated load instantly without permitting the steam pressure at the throttle to drop lower than 750 psig. Motor-generator sets with a modified type of Ward-Leonard control were the only system of fan drives available that would meet these specifications. The turbines are designed to carry rated load on the minimum steam pressure and are provided with a minimum steam-pressure load-limiting device that will cut back on load should the steam pressure fall below this value.

Should an outage occur, the turbine is released automatically from the low block and instantly assumes approximately full rated load. This load is limited by the high-block limit device. The steam pressure in the boiler drum immediately begins to drop. The fans come up to the full speed and the fuel valve opens wide. Fuel is then fired to the boiler until steam pressure is restored to normal for the corresponding load. As soon as equilibrium is restored, the high load limit is moved out of the way and the turbine generator can take load up to its maximum capability.

Results of tests on the system show that the fuel valve opened wide in about 8 sec, and the fans came up to speed in 10 sec. During this time the steam pressure dropped from 1040 psig to 770 psig. The steam pressure then began to rise until full operating pressure was restored in 8 min. It was not possible to duplicate exactly an actual system outage. Hence the load pickup to 48,000 kw instantly, and to 58,000 kw in 40 sec was somewhat slower than would be the case should a system outage occur.

Units 3, 4, and 5 depart from standard practice in one important respect, namely, the regulation of feedwater by controlling pump speed. Feedwater control in units Nos. 1 and 2 is accomplished by means of a conventional throttling feedwater regulator. At low loads this results in extremely high pressures in the feedwater system on the discharge side of the pumps. It also results in a considerable waste of power at all loads. Economic studies indicated that the power saved by installing variable-speed hydraulic drives would justify the additional cost, and expensive boiler-feed regulating valves could be eliminated. Furthermore, intangible savings in maintenance could be effected by eliminating the high excess pressure on piping valves and heaters.

The speed of the pumps is regulated by the boiler's demand for feedwater. This means of control is very stable and no trouble has been experienced in regulating the level of the water in the drums in this manner.

E. H. Krieg.[6] To have constructed such an unusual station as Redondo in less than 18 months would have been a good record under any conditions, but the author has good reason to tell of the achievement so that others may know what can be done.

The design and construction of a power plant on the seashore is indeed a challenge, and the response to that challenge has been unusual in the development of reversal of circulating water for control of mussels and marine organisms in the circulating-water system. The average power-plant operator has a healthy respect for what happens when the flow of water in even small-diameter short-length feedwater lines is stopped even without being reversed, but when the designer and operator must think in terms of 10-ft-diam tunnels that are each 1900 ft long, it takes courage and keen imagination to anticipate the many conditions which can arise to avoid the large losses in investment and outage that anything else but complete success would entail.

No doubt an entire paper could be written on the ingenuity called for in arranging for such quick load pickup, from 5 per cent to full load in 15 sec, particularly when many operators ordinarily try to follow turbine-loading rates of perhaps 5 per cent of rating per min even when no steam-temperature change is involved. Further to plague the designer was the need of providing earthquake protection, not only in the structure but to insure that heavy masses would not start swinging out of phase with other lighter masses to which they are connected. This refers particularly to the intricate problems involved in supporting and damping the vibrations in various parts of a piping system where a relatively light steam line is connected to a heavy nonreturn valve or a heavy turbine emergency stop valve that may weigh 5 or 6 tons.

Now that there are no great hydroelectric developments like Hoover Dam on the horizon to augment system capacity, the author states that future capacity is to be steam-driven. Fuel costs of 30 cents per million Btu are high to one accustomed to thinking in terms of 15 or 20 cents. This naturally develops the thought, expressed by the author, that the reheat cycle has a great deal to offer. This is particularly true of a system which has a great deal of steam-turbine-driven capacity, 20 years old and more, that is so economical for handling short-time peaks overlying a broad base of capacity to be supplied by very efficient units. In other words, where 35 per cent load factor had been commonly associated with steam units, a base is now beginning to develop where steam units in the future will have load factors of 60 per cent and over, even if the full-term life of the unit is taken into account.

High fuel costs point the way toward more efficient units, offering a temptation to go to higher pressures and temperatures. High temperature is the element much more likely to place a limit on quick load pickup which is so important to the system. Reheat is fundamentally more conservative than the higher temperatures for an equivalent heat rate. For example, the reheat cycle is equivalent to approximately 150 F higher temperature in a straight regenerative cycle, and there is little doubt that a 950 F—950 F reheat cycle is more conservative than an 1100 F regenerative cycle at the present time.

When oil is the major fuel, high steam temperatures must be looked at carefully, if not askance, to avoid the very serious wastage of exterior surfaces of tubes caused by vanadium pentoxide which has given so much trouble when used with tubes having external skin temperatures over 1100 F.

The judicious use of partial outdoor construction shows an appreciation of the advantages and also the limitations that such a layout affords. Where salt fogs are possible, protection against condensation in electrical equipment must be provided.

Because Hoover Dam played such a large part in supplying the electricity needs of Southern California during the last two decades, considerable foresight and ingenuity should characterize the decisions to be made on capacity additions during the next decade as they will be mostly of the steam type. Further, the new plants must anticipate burning coal as well as utilizing gas and oil while it is still available. That this challenge is clearly recognized is evidenced by the author's statement: "With an expansion of 760,000 kw in steam-generating capacity either constructed or in progress since the end of the war, the power systems of Southern California are determined that power supply to that area shall be in time and enough."

Author's Closure

Each of the discussions includes many interesting items which were not covered by the author and which add to the value of the paper. Mr. Gilg has described the features of the steam-generating equipment at Redondo most capably. Messrs. Hathaway and Hume have presented excellent discussions of Southern California generating stations whose basic characteristics were determined by the general conditions mentioned in the author's paper, but which have individual features of design which were influenced by local conditions and preferences. Mr. Hathaway has included an able presentation of the arguments favoring the all-indoor station. However, these arguments should be balanced against the savings in construction costs of several dollars per kilowatt which are possible with semioutdoor construction. Mr. Krieg's broad experience in the design of steam-electric generating stations makes his discussion particularly welcome. His comments on the advantages of the reheat cycle are especially pertinent.

The downward trend in Southern California fuel prices, which was mentioned in the paper, has continued. As of October 20, 1949, gas is 16.0 cents per million Btu and fuel oil is 20.6 cents per million Btu. There has been no significant change in the estimated cost of coal.

The increased importance of steam power throughout the Pacific Southwest (Arizona, California, and Nevada) is shown by the change in margin between installed hydro capacity and the maximum demand. Prior to 1940, under favorable water conditions, it would have been possible to carry the maximum demand of the area on the available hydro generation. By 1948, in spite of the addition of nearly 1,000,000 kw of hydro capacity, the demand was 1,400,000 kw more than the total installed hydro capacity.

---

[6] Consulting Engineer, Stone & Webster Engineering Corporation, Boston, Mass. Fellow ASME.

In the Southwest in 1940 only 11 per cent of the power was generated by steam, whereas in 1948 steam power produced 38 per cent of the kilowatthours for the area. The increases in demand since the war, and the further increases which are forecast in all estimates of future demands, indicate increased dependence on steam power for base-load generation throughout California and the Pacific Southwest.

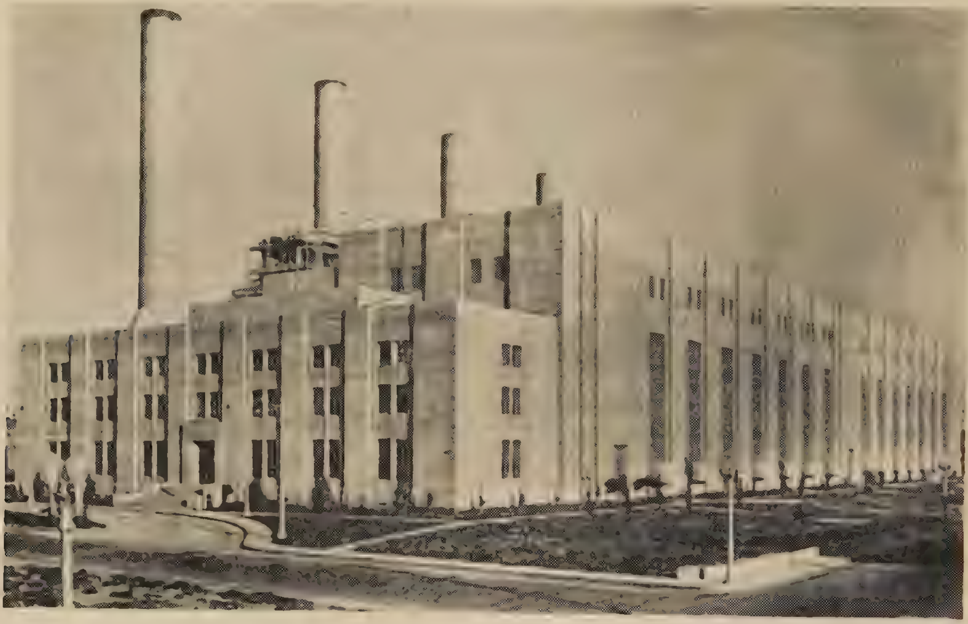


Fig. 9 Completed Redondo Steam Station, Front View

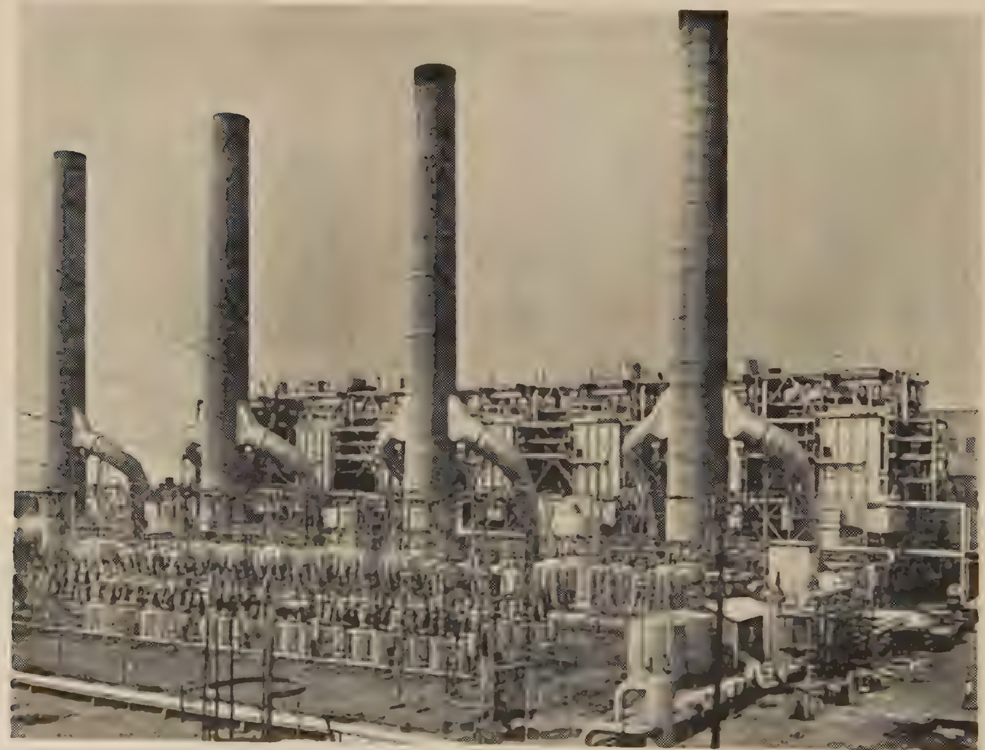


Fig. 10 Completed Redondo Steam Station, Rear View

Postwar Planning for Steam Capacity in Northern California

By C. C. Whelchel[1] and W. R. Johnson,[2] San Francisco, Calif.

The estimated population for northern and central California on January 1, 1949, was 4,950,000. This represents an increase since 1940, of 1,667,000 or about 51 per cent. The Pacific Gas and Electric system peak in 1940 was 1,125,000 kw. In December, 1948, it was 2,050,000 kw under daylight-saving time. Standard time would have added 160,000 kw. If the current trends continue, it is estimated the peak in 1951 will be 2,633,000 kw. Planning the required steam-generating capacity to carry its proper share of this unprecedented load growth is discussed in this paper.

Introduction

The northern and central part of California served by the Pacific Gas and Electric Company comprises an area of approximately 90,000 square miles with a population approaching 5,000,000. Along with other sections of the West Coast, this area is experiencing an unusual postwar growth in population, in industry, and in the general level of business and agricultural activity. This growth is reflected in the increasing electric power load, and by the current construction program of the authors' company, under which nearly 1,300,000 kw of new hydroelectric and steam-electric generating capacity will be added to the resources of the area before the end of 1951.

The purpose of this paper is to present the background leading up to this current expansion program, to discuss reasons for the location and amounts of steam capacity being added to the system, and to explain unusual technical features of these new plants.

Load Growth and Power Supply

The territory served with electric power largely by the Pacific Gas and Electric Company is a naturally coherent geographical area nearly 500 miles in length, and between 150 and 200 miles in width. The heart of this region is the great valley of the Sacramento and San Joaquin Rivers, bounded by the Sierra Nevada on the east, by the coast ranges on the west, and by other mountain groups on north and south. The natural outlet for the valley, both geographic and economic, is through the San Francisco Bay region.

In Fig. 1 certain features of this territory are shown, including its main watercourses, and location of the company's hydro and steam-electric power plants. The hydro plants, with minor exceptions, are scattered along the western slopes of the Sierra Nevada. They are located on many different rivers, progressing from the Pit in the north, down along the Feather, Yuba, Bear, American, Mokelumne, Stanislaus, San Joaquin, Kings, and finally the Kern River in the extreme south. Hydro developments by others are not indicated; notable among these omissions are the Big Creek development on the San Joaquin by the Southern California Edison Company, Ltd., the Shasta and Keswick powerhouses of the United States Bureau of Reclamation, the City and County of San Francisco plants on the Tuolumne, and various irrigation district projects.

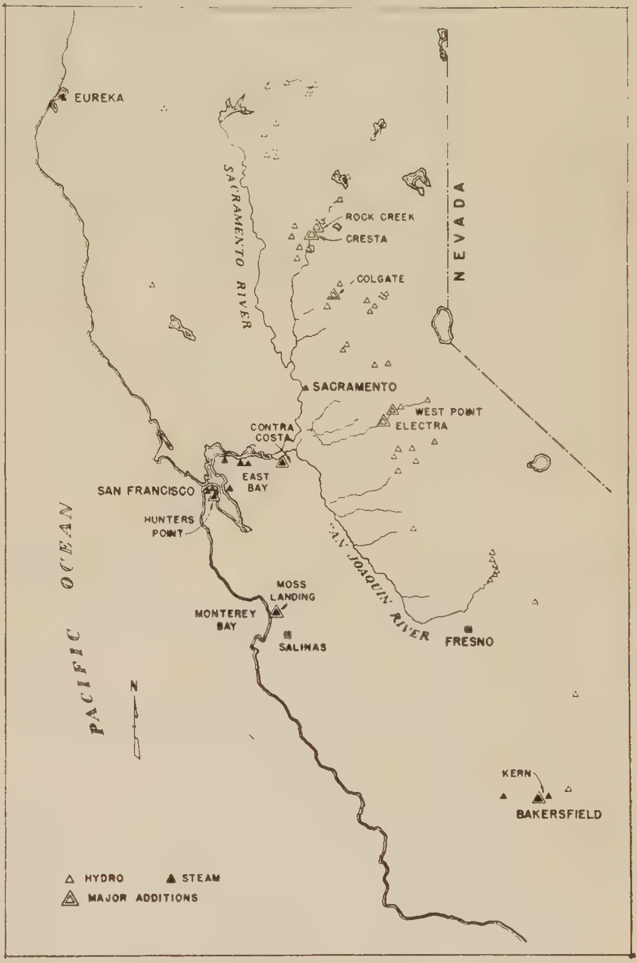


Fig. 1 Territory Served by the Pacific Gas and Electric Company

(The scale for this map is roughly 1 in. to 100 miles.)

Steam-electric plants are indicated by solid triangles. They are located in the San Francisco Bay region, along the coast, and in the extreme southern part of the territory. Plants built, or under construction, in the postwar expansion program are indicated by a double triangle.

After a brief decline in the early 1930's, the consumption of electric energy has climbed steadily upward. This is shown by the curves of annual peak kilowatts and energy in Fig. 2. During the war years growth was accelerated. As the peak of war activity passed there appeared to be a slight decrease in the rate of growth, particularly in 1945 and 1946. It was commonly anticipated, by business and government alike, that a return to prewar

---

[1] Chief, Division of Steam Engineering, Pacific Gas and Electric Company. Mem. ASME.

[2] Division of Hydro-Electric and Transmission Engineering, Pacific Gas and Electric Company.

Contributed by the Power Division and presented at the Semi-Annual Meeting, San Francisco, Calif., June 27–30, 1949, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 49—SA-10.

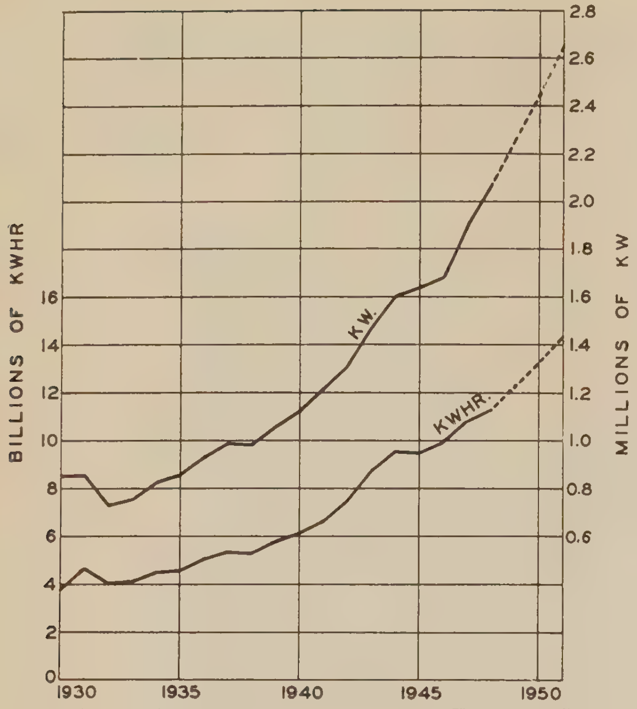


Fig. 2 Annual Peak Kilowatts of Load and Energy of Pacific Gas and Electric Company System

trends would occur. The incorrectness of this reasoning is clearly indicated by the load growth experienced in 1947 and 1948. The system peak of 2,078,900 kw in 1948, represented an increase of 9 per cent over the 1947 peak; while 1947, showed an increase of 13 per cent over 1946.

From 1940, the last prewar year, to 1947 inclusive, peak demand increased from 1,125,700 kw to 1,906,200 kw. This increase of 780,500 kw in 7 years is a growth of 69 per cent, or an average yearly increase of approximately 8 per cent.

In this same period the Pacific Gas and Electric Company brought 206,000 kw of hydro capacity and 100,000 kw of steam capacity into operation, a total addition of 306,000 kw. Also the United States Bureau of Reclamation brought into operation two units at Shasta Dam with a name-plate rating of 150,000 kw. The additions by the power company were projects well started and equipment largely manufactured prior to the war. Exceptions were 32,000 kw of peaking capacity at Pit River No. 1, obtained by enlarging forebay capacity for existing units, and two small steam units in the Eureka area in 1946–1947. Orders for two large steam units placed prior to the war in 1941, were held in abeyance in the interest of the over-all national defense program. These orders were reinstated at the end of the war, and both units went into operation during 1948—seven years after the original orders were placed.

In addition to the two large steam turbogenerators and the smaller installations, just mentioned, planning for other new generation was pursued vigorously as the war period drew to a close. The program, however, faced serious delays beyond the company's control. Manufacturers found themselves swamped with orders for major equipment items; the wave of labor difficulties delayed production. Furthermore, the vagaries of nature brought subnormal rainfall during 1947, followed by the driest January and February, on record in 1948.

In spite of the most strenuous efforts by the company, the effects of drought in reducing hydrogeneration and greatly increasing the agricultural irrigation pumping load, made necessary a power-curtailment program. This period was of short duration owing to the combined effects of daylight-saving time, heavy spring rains, power delivered from southern California, and completion of new plants in northern California. The new plants brought into operation at this time were the first major units completed in a postwar program of considerable magnitude.

The Postwar Program

A construction program of great magnitude is now well under way to restore adequate power margins and to serve the rapidly growing needs of the area. The estimated net generation capacity increases, constructed and planned for the 1946–1951 period, are given in Table 1.

TABLE 1 ESTIMATED NET GENERATION CAPACITY INCREASES—1946–1951

	Steam, kw		Hydro, kw	
1946–1947	Two additional units, boilers and improvements to existing plant in the Eureka area (approx).......	12000	Pit River No. 1 forebay peaking increase on existing units	32000
1948	Kern No. 1..........	75000	New Electra[a]	87000
	Hunters Point........	100000	West Point	13000
1949	Hunters Point.......	100000	New Colgate[a]	23000
			Cresta	68000
1950	Kern No. 2..........	100000	Rock Creek	110000
	Moss Landing........	100000	Potter Valley—peaking increase on existing units...........	4000
1951	Moss Landing........	100000		
	Moss Landing........	100000		
	Contra Costa.........	100000	Total hydro..........	337000
	Contra Costa.........	100000		
	Contra Costa.........	100000		
	Total steam..........	987000		

[a] Replaces existing generation at the two plants totaling about 30,000 kw.

This program was decided upon after consideration of many factors relating to the amounts of steam and hydro capacity to be added. The basic reasons for the large proportion of steam generation include the following:

1 Steam-capacity additions postponed by the war. The first units at Kern and Hunters Point are in this category. These steam-capacity additions had been planned prior to the war as necessary to maintain the desirable steam-hydro ratio in the area.

2 Steam capacity required because of the changing economic steam-hydro ratio. The economic and other factors entering into the optimum proportions of steam and hydro capacity have been discussed fully in earlier papers. As the more favorable hydroelectric sites are developed,[3,4] those remaining are more costly. At the same time technological improvements in steam-plant design, such as larger unit size and higher efficiency, have tended to reduce the relative cost of steam. Offsetting this has been a rising fuel cost. However, the over-all result has been a gradual shifting of the economic steam-hydro ratio from approximately 75 per cent hydro and 25 per cent steam to nearly equal amounts of each.

This trend is illustrated in Fig. 3, which shows the annual energy sources for the company's system from 1930 on, in per cent of the total for each year. The energy shown coming from "other sources" is almost entirely from hydroelectric plants owned by others. The proportion of energy from fuel plants is as low as 3 per cent in 1935, and is seen to be increasing gradually, reaching 36 per cent in 1947. However, the trend is partially obscured by the effect of variations in precipitation, both 1947 and 1948 being below average. In such years the amount of steam energy is much greater than normal.

3 Steam capacity for support of hydro developments planned by the company after 1951.

---

[3] "Aspects of Steam Power in Relation to a Hydro Supply," by A. H. Markwart, *Mechanical Engineering*, vol. 48, 1926, pp. 557–561.

[4] "Mingled Hydro and Steam Power Production in California—Past, Present, and Future," by A. H. Markwart, Trans. AIEE, vol. 59, 1940, pp. 775–781.

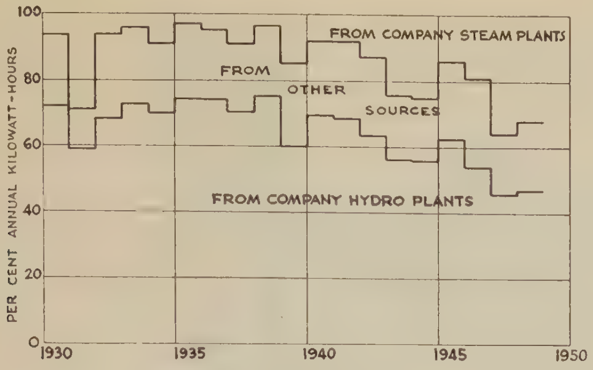


Fig. 3 Energy Sources for the Pacific Gas and Electric Company System

4 Factors in planning and construction. Nearly all new hydro developments require procurement of licenses and water rights involving Federal agencies and other interests. Until satisfactory agreements are reached on these matters, design and manufacture of the major equipment cannot be commenced, nor preliminary construction begun.

Steam turbogenerators, on the other hand, do not have their design dictated by the site where they are to be installed. Therefore, orders may be placed and design commenced far in advance of the procurement of a satisfactory site.

5 The company has always followed a policy of planning to facilitate the absorption of hydroelectric energy produced by irrigation districts, the United States Bureau of Reclamation, and other agencies as a by-product of water-conservation projects. The prime importance of development and best use of California's water resources have long been recognized. The company endeavors to co-operate to the utmost in the utilization of incidental power from these projects for the greatest possible benefit to the project.

System Influences on Steam-Plant Locations

With some artistic liberties, the problem of load and power supply on the company's system for an estimated average water year as of 1951, is shown in Fig. 4. Energy from hydroelectric sources is represented by the unshaded arrows, that which must be supplied from steam plants by shaded arrows, and the quantity of energy involved is proportional to the arrow width. It is rather apparent that major steam plants should be located in the southern and west-central portions of the area if transmission losses are to be minimized.

Planning for steam plants of the capacities needed for this program required consideration of factors not involved in smaller installations located in metropolitan areas where power may be fed directly into the distribution network. It is not feasible on the company's system to take away the full output of a large plant entirely over distribution voltage circuits, nor can the large blocks of generated power be absorbed locally at periods of light load.

In view of the foregoing the new plants were designed for system, as well as local area use, and it was necessary in each case to provide full-capacity outlets directly to the transmission system. With such outlets the general plant locations were determined after considering such factors as area power supply and load, availability of fuel, make-up water, condenser circulating water, foundation conditions, transmission outlets, and the possibility of future expansion.

All four of the new large steam plants required construction of high-voltage transmission connections. This was accomplished for Hunters Point by means of 115-kv underground-cable connections to a transmission substation in combination with power supply to a new 115-kv distribution substation in downtown San Francisco. The Kern plant connects directly to the 115-kv transmission by overhead lines to substations in the vicinity, and is provided with transformer auto taps to feed into the 70-kv subtransmission. At Moss Landing two of the units will have auto tap connections tying into the 115-kv transmission lines, in the area, but all three units will have full-capacity outlet by 230-kv lines to the San Joaquin Valley and the lower San Francisco Bay regions. While the other plants have connections to the 115-kv or subtransmission, the entire output of the Contra Costa plant will be transmitted over the 230-kv high-voltage network.

Kern Steam Plant[5,6]

Agriculture has long been recognized as the basic industry in California and the one which in a large measure is responsible for the prosperity of the state. The San Joaquin Power Division of Pacific Gas and Electric Company, with headquarters at Fresno, represents over 70 per cent of the system's agricultural load. In 1946, the peak for this division was 415,000 kw. In 1948, it was 578,000 kw. The estimate for 1951, based upon present trends is 800,000 kw, which represents a 93 per cent increase above 1946, a 5-year period.

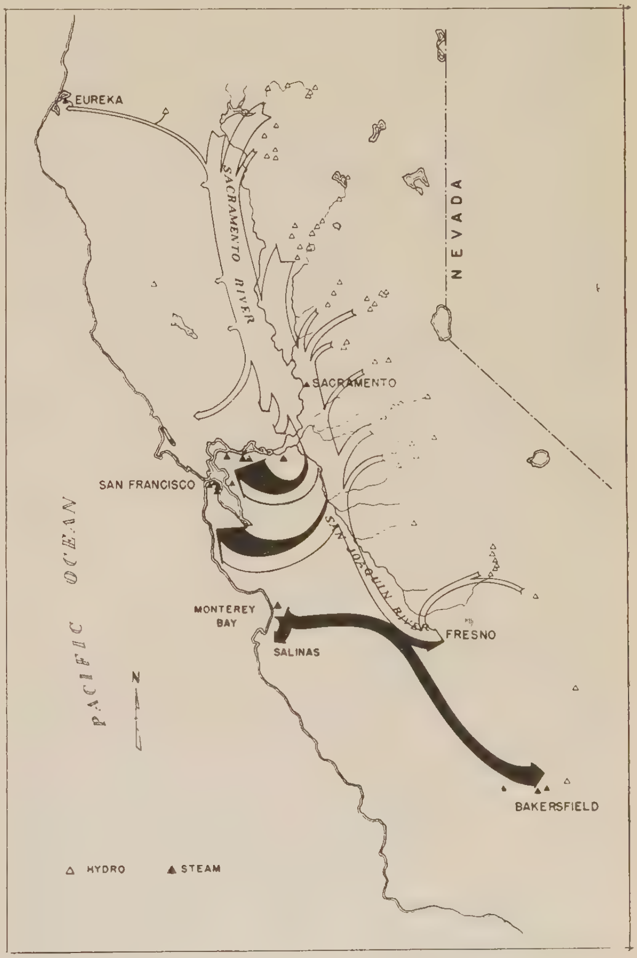


Fig. 4 Estimated Load and Power Supply for the Pacific Gas and Electric Company System in 1951, With an Average Water Supply
(The scale for this map is roughly 1 in. to 100 miles.)

---

[5] "Control is Centralized at Kern station," *Electrical World*, vol. 130, Sept. 25, 1948, pp. 90–91.

[6] "Centralized Control at Kern," *Electrical West*, vol. 101, 1948, pp. 79–80.

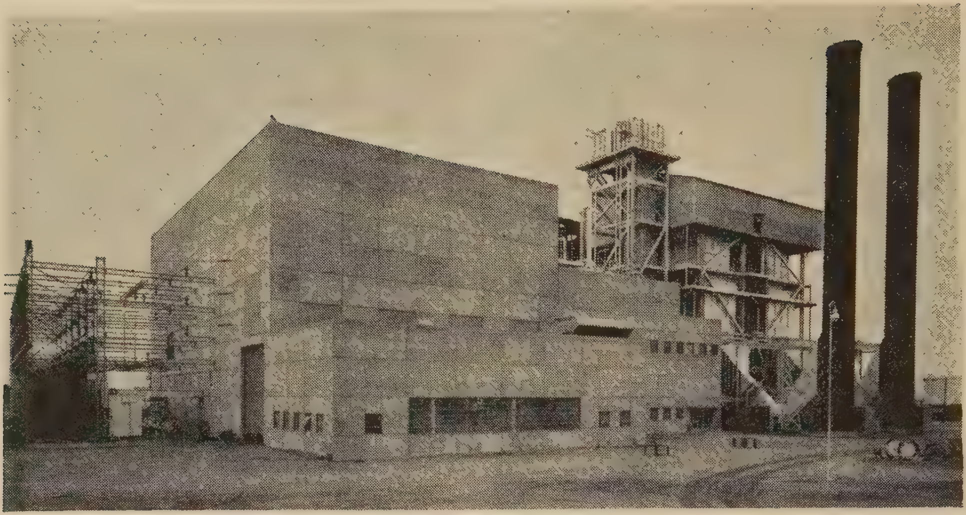


FIG. 5 KERN STEAM PLANT

In the lower end of this territory, power had been supplied from transmission sources, small hydro and steam plants, and from an interconnection with the Southern California Edison Company at Magunden. Upon termination of the Edison Company's interchange agreement in December, 1947, a replacement of this power source was required. Additional steam capacity in the Bakersfield area proved to be an economical means of replacing this supply, as well as providing for a portion of the load growth in the lower San Joaquin Valley. Since there are no rivers in the area capable of supplying condenser circulating water, cooling towers must be used, and their make-up is supplied by wells. The use of cooling towers permitted the plant to be located on a level 160-acre plot near oil refineries which supply fuel oil by pipe line. To those not accustomed to obtaining condensing water in this manner, it may at first seem strange to locate a large steam power station in an open field. That, however, proved to be most suitable for this station which is 5 miles west of Bakersfield on the Santa Fe Railroad.

The first main unit at Kern, which is rated at 75,000 kw, has been in commercial operation since May 10, 1948. A 100,000-kw second main unit is scheduled to be placed in operation during January, 1950. There will be two accompanying house units of 6000 and 7500 kw, respectively. Each of the four boilers has a normal full-load evaporation rate of 450,000 lb of steam per hr. These boilers are designed to burn either fuel oil or natural gas. A general view of this plant is shown in Fig. 5.

MOSS LANDING STEAM PLANT

Another important agricultural center, sometimes called the "Salad Bowl of America" because of its tremendous production of lettuce, celery, and artichokes, is located along the coast of Monterey Bay, with much of the business activity centering at Salinas. The agricultural lands here must also be irrigated by water pumped from wells. In this area at Moss Landing, a 270-acre site was selected as the location of a new plant in which three 100,000-kw main units and three 7500-kw house units will be installed initially. As in the other stations, two boilers per turbine have been provided. An appreciable amount of the power from this plant will be used locally with much of the remainder normally being transmitted east over the coast range mountains to the San Joaquin Valley in the general areas of Fresno and Bakersfield.

Condenser circulating water will be drawn from Moss Landing Harbor and discharged into Elkhorn Slough to the north. Since Elkhorn Slough is connected to the harbor, a careful study of the possible effects of recirculation was undertaken. It was concluded that while certain conditions of unfavorable tidal flow may result in some temperature elevation, this will occur infrequently and should not affect the capacity of the first three turbines. Operating experience with the initial installation will determine what, if any, further precautions must be taken when additional units are installed. Fresh water for the station will be obtained from wells located on the property.

Fuel oil will be delivered by 120,000-bbl tankers through a submerged pipe line extending approximately 3500 ft out into Monterey Bay. A flexible hose at the sea end of this line will be taken aboard the tanker upon arrival and connected to the discharge of its unloading pumps. When not in use, the hose will rest on the bottom of the sea. If conversion to coal operation should become desirable at this station, a spur of the Southern Pacific Railroad will be available for coal deliveries. There is ample space at the plant for the simultaneous storage of fuel oil and coal. Grading at the site was started in September, 1948.

"STATION P," HUNTERS POINT[7]

Load growth in San Francisco and on the Peninsula necessitated an increase in power supply for this area. Studies indicated that this could be obtained most economically by installing additional steam capacity at Station P where space was available and a good condenser circulating-water system had already been developed. The original unit at this station was placed in operation in 1929, and is rated 35,000 kw.

The two new 100,000-kw main units and two 7500-kw house units are supplied steam by four new boilers entirely independent of the old unit so far as operation is concerned. Circulating water is drawn from San Francisco Bay. Fuel oil is delivered by barge or tanker to a filling line carried on a pier extending into the bay.

Power generated by the new units is stepped up to 115 kv and transmitted by high-voltage underground cable to the new Mission substation in the heart of the city. Two 115-kv underground cables also connect Station P with Martin substation from where the power may be taken during light load periods over transmission lines to the system outside the city.

The first new unit at Station P was placed in commercial operation on December 12, 1948, and the second followed on January 25, 1949. Construction started in July, 1947, and about 18 months were required to bring the first unit into operation.

---

[7] "Station P," *Power*, vol. 92, 1948, pp. 586–588.

CONTRA COSTA STEAM STATION

The most concentrated industrial area of the company's system lies in the East Bay region and along the lower Sacramento and San Joaquin Rivers near where they empty into San Francisco Bay. The Contra Costa Station has been located in this area at a point $2^1/_2$ miles east of Antioch on the San Joaquin River.

The backbone of the electrical transmission system is the 230-kv network running the length of the great Central Valley and carrying power from the many hydroelectric plants in the Sierra Nevada to the principal load centers in the San Francisco Bay area and below. The Contra Costa steam plant was placed astride this network. Its power will be used wherever required in the system, and its location on the San Joaquin River was, among other considerations, decided on the basis of availability of circulating water and proximity to fuel-oil supplies along the upper bay. Circulating water will be drawn from the river which, although subject to tidal action is still essentially fresh. It is believed that stratification and the large volume of water available will reduce the effect of recirculation during unfavorable conditions of tidal and stream flow. Fuel oil will be delivered by barge, tanker, or pipe line. Coal when and if required in the future can be delivered to a spur of the Santa Fe Railroad entering the property.

The initial installation at this plant will include three 100,000-kw main units, three 7500-kw house units and six steam generators. All units are scheduled to be in operation during the first 6 months of 1951. Construction was started in February, 1949.

POWER-PLANT DESIGN AND STATION EQUIPMENT

The steam program outlined in this paper has been and is being planned and co-ordinated by the Pacific Gas and Electric Company's engineering department but, except for the units at Eureka, the plant design and the construction are being carried out by others. Stone and Webster Engineering Corporation has this responsibility for Kern, Station P, and Moss Landing. The Bechtel Corporation has similar responsibilities in connection with the Contra Costa plant.

Structure. In general, each of the four new steam stations is of semioutdoor design, with the turbine room and control bay entirely housed. The boilers are protected by an umbrella-type roof of corrugated transite. The firing aisle is completely enclosed. The buildings are of steel-frame construction with reinforced-concrete walls and roofs. A plan and section of the Contra Costa steam plant are shown in Figs. 6 and 7.

Turbine Generators. Except for the two 1800-rpm Kern units, where delivery was a prime consideration, the remaining eight main turbine generators involved in this postwar program, each with a nominal rating of 100,000 kw, 15 lb hydrogen, are 3600-rpm machines designed for 1300 psig and 950 F. The turbines are of the tandem-compound double-flow condensing type and have four extractions to supply steam to the feedwater evaporators and the closed feedwater heaters. The condensers are single pass with divided water boxes to permit part-load operation on one of the two circulating-water pumps.

The eight accompanying automatic-extraction house turbogenerators are rated 7500 kw and are designed for the same steam conditions as the main units. They operate condensing with a single extraction opening supplying steam for the deaerating feedwater heater. The house units are installed to provide an independent electric power supply for the vital station auxiliaries. They are simple rugged units expressly designed immediately to assume full load in the event of a system disturbance. Normally they will be in operation supplying extraction steam at a constant 5-psig pressure to the deaerating feedwater heaters. System frequency or voltage dips, severe enough adversely to affect station operation, automatically disconnect the main units from the house bank. Thus, the main units will operate at whatever frequency and voltage the system can develop and will carry essentially full load. Meanwhile the house units furnish normal voltage and frequency to the auxiliaries.

A heat-balance diagram, which is typical of each of the eight main and house units at Station P, Moss Landing, and Contra Costa is given in Fig. 8, for the 100,000-kw load condition.

Steam Generators. All steam generators included in this program are of the radiant type with tangent waterwall tubes in the furnace. Two different methods of superheat control are employed. At Contra Costa and Moss Landing, a portion of the gases are by-passed around the primary superheater, and the position of the by-pass dampers determines the temperature of steam leaving the superheater. At Kern and Station P, the control is by means of a heat exchanger installed in an enlarged lower drum. Part of the steam leaving the primary superheater is passed through this attemperator and is cooled prior to mixing with the remainder. All steam then passes through the secondary superheater and thence to the turbines. Two different types of air heaters are also employed. The first eight boilers in the program are equipped with tubular air heaters, whereas the last twelve use the regenerative type, wherein the heat-exchange elements are exposed alternately to furnace exhaust gases and to incoming air.

The steam generators for Station P and Kern were designed for oil and gas fuels. Conversion to coal, should this become desirable, can be accomplished with some loss in capacity. At Moss Landing and Contra Costa, on the other hand, provision is being made for possible use of coal with full-capacity output. Not only is space being provided for installation of such auxiliary equipment as pulverizers, coal bunkers, and dust eliminators, but the steam generators themselves are provided with larger furnaces to permit burning coal efficiently and without reducing the steaming rate. The difference in furnace size is illustrated by the heat-release rate, which for Kern and Station P is about 30,000 Btu per cu ft-hr on oil at rated load. Heat release for the same conditions at Moss Landing and Contra Costa by contrast is approximately 18,000 Btu per cu ft-hr.

In addition to providing greater volume to permit the slower-burning coal to be consumed before the gases leave the furnace, greater space between tubes in the superheater is provided to minimize effects of slagging of these surfaces. The minimum clear space between tubes in the Kern and Station P superheaters is 1 in. The Moss Landing and Contra Costa superheaters have 2 in. clear. These more conservatively designed boilers are expected to be able to stay on the line longer when burning oil fuel and thereby improve boiler reliability.

Quick Pickup. All of the new steam stations at times will be operated at low loads on stand-by duty and may then be called upon to provide nearly full-load output in a matter of 20 to 30 sec. This quick-response requirement is reflected in the selection of equipment. Boiler feed pumps and draft fans are operated always at full speed immediately ready for full-capacity demand. This is achieved at some loss in cycle efficiency because of the need to resort to dampers, throttling valves and recirculation of boiler-feed-pump flow. The fuel burners and auxiliary equipment have a capacity range of 20 to 1 at Kern and Station P, and 10 to 1 for Moss Landing and Contra Costa. All burners are always in service.

Centralized Control. Centralized control is a basic feature of the general plant layout of each of the stations and the operator at the key point in the control room is in position to maintain visual contact with helpers operating the principal items of equipment. He is able to see the main turbines, house units, and boiler feed pumps on the one hand, and the boiler firing aisle on the other. In addition, he is aided in the operation of the plant by a very spe-

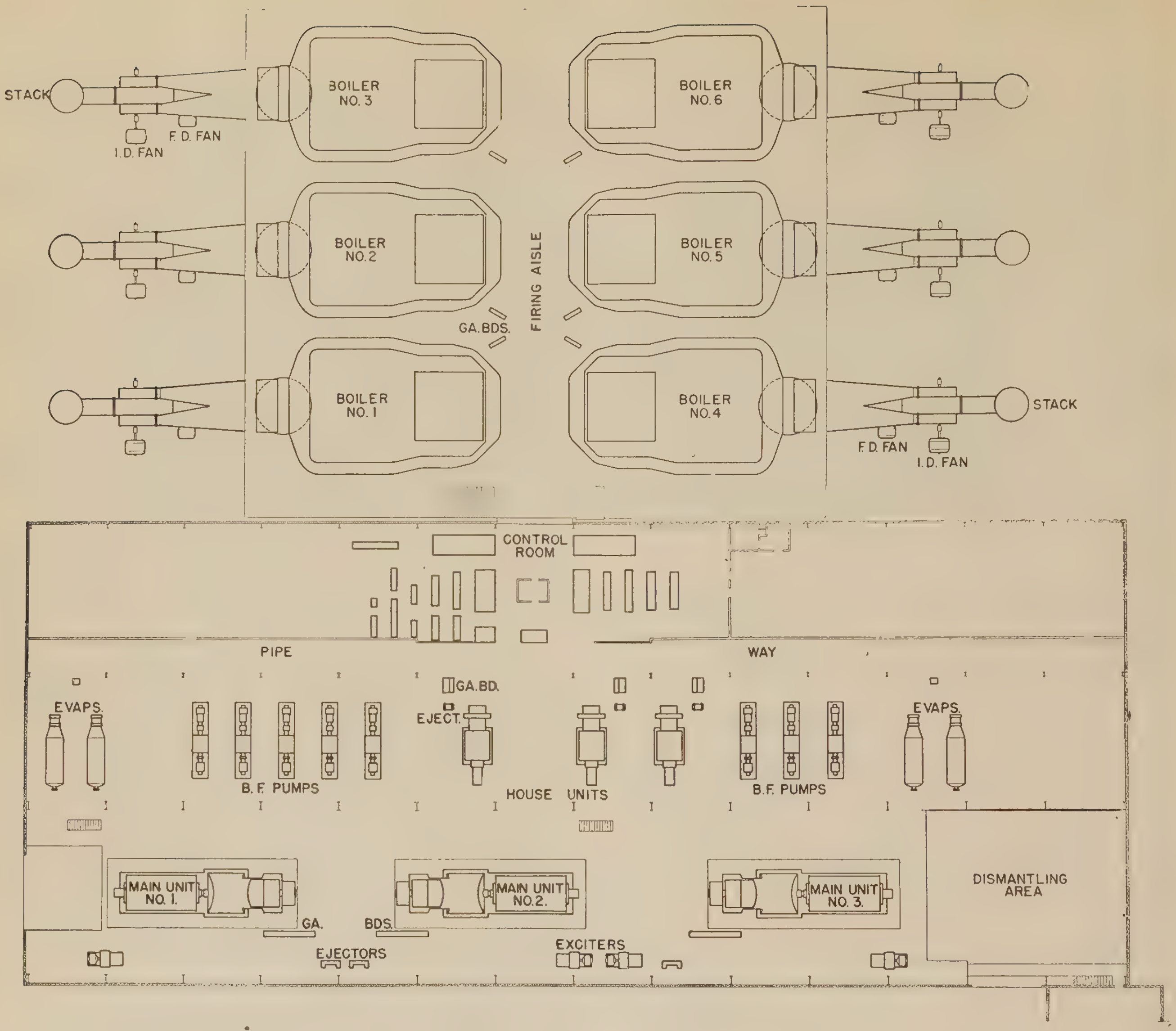
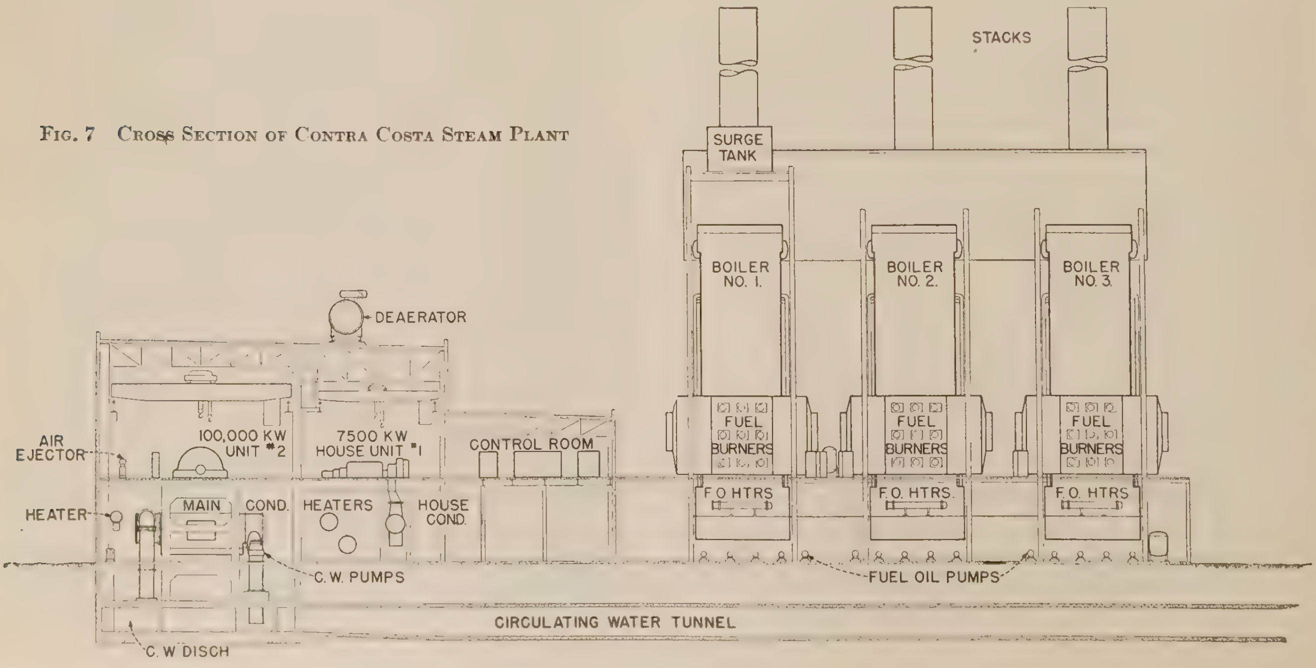


Fig. 6 Plan of Contra Costa Steam Plant

Fig. 7 Cross Section of Contra Costa Steam Plant



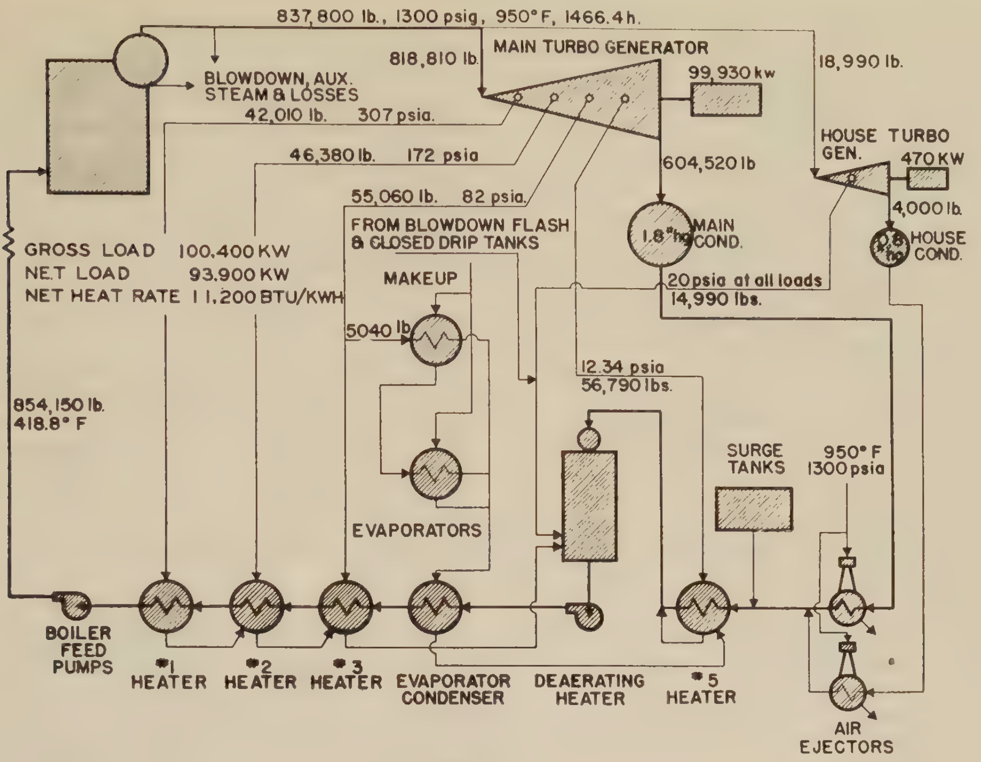


FIG. 8 TYPICAL HEAT-BALANCE DIAGRAM

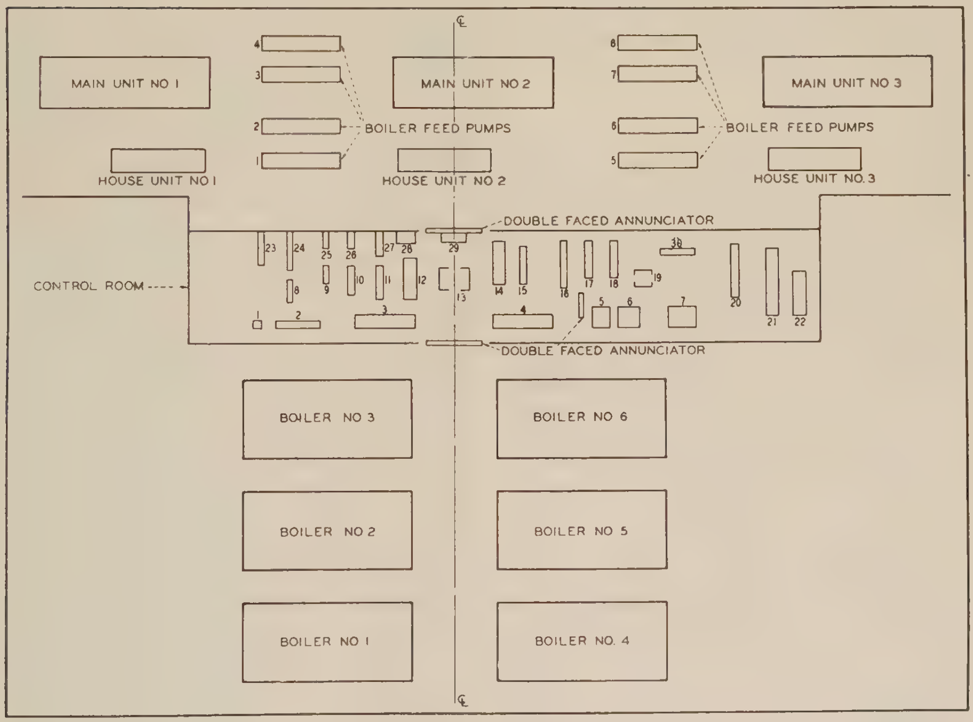


FIG. 9 CONTROL ROOM AT MOSS LANDING

1 Control air-compressor auxiliary board.
2 Fuel-oil control board.
3,4 Individual boiler control boards.
5 110-Kv control board.
6 Generator and transformer banks.
7 220-Kv control board.
8 Fuel-oil heaters control board.
9 Miscellaneous heaters control board.
10 Ratio control board.
11 Feedwater meter board.
12 Master control board.
13 Plant operator telephone booth.
14 House-generator control boards.
15 House-turbine auxiliary control boards.
16 Load-frequency control board.
17 4.16-Kv control board.
18 480-Volt control board.
19 System operator telephone booth.
20 Excitation boards.
21,22 125-Volt d-c station service switchboards.
23 Soot-blower reducing-valves control board.
24 Miscellaneous recording meter board.
25 Miscellaneous pumps gage board.
26 200-Lb reducing-valves gage board.
27 Feedwater-pump gage board.
28 Feedwater control board.
29 Main generator control boards.
30 Electrical recording meter board.

cial annunciator system consisting of banks of large light-type indicators mounted above the various control boards and on the control-room walls between the turbine and boiler rooms. By a combination of horns and signal lights, he is warned of abnormal operating conditions anywhere in the plant. By use of a color code, the relative urgency of corrective measures is emphasized.

At Kern and Station P two main units and their auxiliaries are controlled from one room. At Moss Landing and Contra Costa three main units will be controlled from one room. A block diagram of the Moss Landing Station with identification of the individual control boards is shown in Fig. 9.

Station Auxiliary Power System. The primary auxiliary power system in each of the stations is 4160-volt 3-phase. The switchgear used therewith is metal-enclosed with magnetic-type air breakers. The buses are connected in the form of a ring, one ring bus for each main generating unit. Each bus has connected to it, a house turbine and a house transformer bank. The bus sections are connected together through a bus tie switch, allowing parallel operation if necessary.

The secondary auxiliary-power-system voltage is 480 volt 3-phase, and is a selective radial type consisting of load centers located at strategic locations around the plants. Each load center is equipped with a number of 3-pole electrically operated air breakers of proper interrupting duty. The main load centers are supplied with power from 4160/480-volt stepdown transformers. These main load centers supply power to other load centers scattered around the plant. Each center is sectionalized so that only one half the equipment is lost in case of a bus failure.

Motors of 250 hp and above are connected to the 4160-volt system, while smaller motors are connected to the 480-volt system. In addition, the power for the lighting system is 3-phase, 4-wire, 216/125-volt, supplied by numerous small 3-phase lighting transformers stepping down from 480 volts to 216 volts.

Conclusion

The construction program outlined in this paper covers almost 1,000,000 kw of new steam-electric generating capacity which will be added to the area before the end of 1951. This amount of capacity clearly indicates the increased importance of fuel-burning plants to the Pacific Gas and Electric Company. From a system standpoint, the new steam stations not only add generating capacity, but the required interconnecting transmission lines and switching stations will further contribute to system reliability. As integration and growth continues, the need for "quick pickup" steam stations and multiple-boiler per turbine installation probably will tend to decline. The future trends of steam-station design here as well as elsewhere is expected to be in the direction of simplification of equipment and plant layout and still lower station heat rates.

Discussion

M. W. Carty.[8] The authors give a clear presentation of the salient factors considered in providing additional electric power supply for the requirements in the great central-valley area of California. The rapid rate of growth in peak demand in that area is probably unprecedented in the United States for any comparably sized territory. It is of interest to note that the economics of the situation for the extensive territory served will lead by 1951 to a 50-50 per cent ratio of steam and hydroelectric power-generating plants. In the power developments recently completed since the end of the war in this area, or now under construction, the added steam capacity will be three times as great as the added hydro power.

Fig. 4 of the paper shows clearly the preponderant dependence on hydroelectric power in the northern part of the system and makes evident the strategic value of added steam capacity in the San Francisco Bay area. It is a wise precaution to locate the large steam plants where fuel oil is readily available by tanker delivery as the quantities of fuel are large. The Moss Landing steam plant will produce a net output of 565 kwhr per bbl of oil when operating under full-load conditions, and approximately 12,800 bbl of oil will be required per day under these conditions. If the plant is operated at the minimum capacity at which the turbines will be capable of quick pickup of load, approximately 1000 bbl of oil will be burned per day without any net output of energy.

The steam plants have multiple boilers per main turbine-generator and this was decided upon, having in mind the requirements of quick pickup of load, the need to remove from time to time metallic adhesions on the furnace tubes resulting from oil burning, and the need to wash off occasionally deposits on the superheater tubes. Satisfactorily large water storage in the boiler steam drums is helpful to give accumulator action at times of quick pickup of load, and multiple boilers are better in this respect than single equivalent large boilers. In a completely integrated and compact system, as distinct from the central-valley area with its long transmission lines, it would be desirable to come closer to the "unit principle" of one boiler per turbine generator without cross-connections, in the interest of lower investment cost and operating simplicity.

The advantages of centralized control are apparent, and this has been a major factor in the general arrangement of the steam plants. The operators have direct vision from the central operating point to the key centers at the turbine generators, boiler feed pumps, and house turbines.

Referring to Fig. 5 of the paper, which shows the Kern steam plant, it is not entirely apparent that the two principal operating levels, main operating floor and ground floor, are each arranged at one continuous uniform level which adds to the convenience of the operators and ease of maintenance activities. This plant is dependent upon cooling towers for cooling the condenser circulating water. For make-up and other requirements at full-load operation, the water requirement is at the rate of 4000 gpm, which is readily obtained from deep wells on the power-plant property.

Fig. 9, the Moss Landing steam plant, shows the boiler-feedwater pumps located on the main operating level, and consequently they are always under the observation of an operator. The house units also are located in readily observable positions and are easily accessible. Fuel oil for this station will come from tankers having a capacity of 120,000 bbl and will be discharged by the tanker pumps operating at 125 psig pressure and will be heated to 140 F, if necessary, in the tanker before delivery. It is expected that the tanker will be able to deliver up to a maximum rate of 8000 bbl of oil per hr through an 18-in. submarine pipe and a 24-in. on-shore pipe, extending to the storage tanks at the steam plant, a distance of approximately 1$^3/_4$ miles.

Arrangements in the steam plants are made with a view to the safety and comfort of the operators, and primarily for reliability of service.

Prevention of trouble from mussel accumulation in the circulating-water lines has received special attention at the Moss Landing steam plant. There are two supply lines to each main condenser, and it is planned to circulate water warmed to 95 F back through one line at a time. The single-pass divided-water-box arrangement of the condenser is well-adapted for this operation. A routine backwash may be established for one condenser circuit per week end, resulting in a normal 3-week time cycle. Perhaps a longer cycle will suffice, and the usual chlorine concentration for reduction of slime formation inside condenser tubes may be increased during the backwashing operation.

---

[8] Senior Mechanical Engineer, Stone & Webster Engineering Corporation, Boston, Mass. Mem. ASME.

C. H. Woolley.[9] The Pacific Gas & Electric Company is very progressive and has consistently purchased efficient and modern designs of power-generating equipment in order to best serve the rapidly growing population. In 1938 the company with which the writer is affiliated installed the first oil-fired radiant-type boiler for P.G.&E. The single-pass principle of gas flow which results in low fan-power costs and improved operation was incorporated in this design. When these boilers were designed, the serious fouling of superheaters with oil slag had not yet been encountered in California; however, the front row of superheater tubes was staggered to form a 3-row slag screen.

When the next expansion of this company was contemplated in 1941, for Kern steam plant and Hunters Point, the same modern radiant-type boiler was selected, but at that time the oil-slag problem had been encountered, and special provisions were included in the design to reduce the possibility of an outage due to the effect of oil slag. The furnace was changed from the older stud-tube design to touching-tube design, in order to provide a cooler furnace and to eliminate the growth of furnace walls, caused by oil slag penetrating the refractory on the studded tubes, as reported by V. F. Estcourt.[10] The tubes in the first two sections of the superheater were spaced far apart so that oil slag could not close the lanes. The superheater banks were made shallow in order to permit better cleaning by the soot blowers. Provision was made to water-wash the superheater, economizer, and air heater. Drain hoppers were installed under all three of these units of equipment.

Unfortunately, the war delayed the installation of the boilers, and they were not placed in operation until 1948.

When it was necessary to purchase the boilers for Moss Landing and Contra Costa, there was not sufficient operating experience to determine if the same improvements would render the units free from the increasing oil-slag troubles. As the degree of trouble due to this slag had increased throughout the country, it was believed that even greater precautions should be taken to insure continuous operation. The possibility of future coal-firing required that the furnace for the new stations should be larger, and this automatically gave a lower temperature entering the tube bank with oil-firing than would have been used normally for a straight oil-fired job. Fig. 10 of this discussion shows the boilers for the Moss Landing Steam Plant, with which the writer is familiar. The additional precautions against tube fouling were as follows:

The front bank of superheater tubes were placed on 12-in. centers primarily to guard against coal-slag troubles, but automatically, operation became safer for oil-firing. As the gas temperature was reduced and reached the second superheater section the spacing was changed to 6-in. centers which is adequate in this temperature zone to insure against tube-lane pluggage with

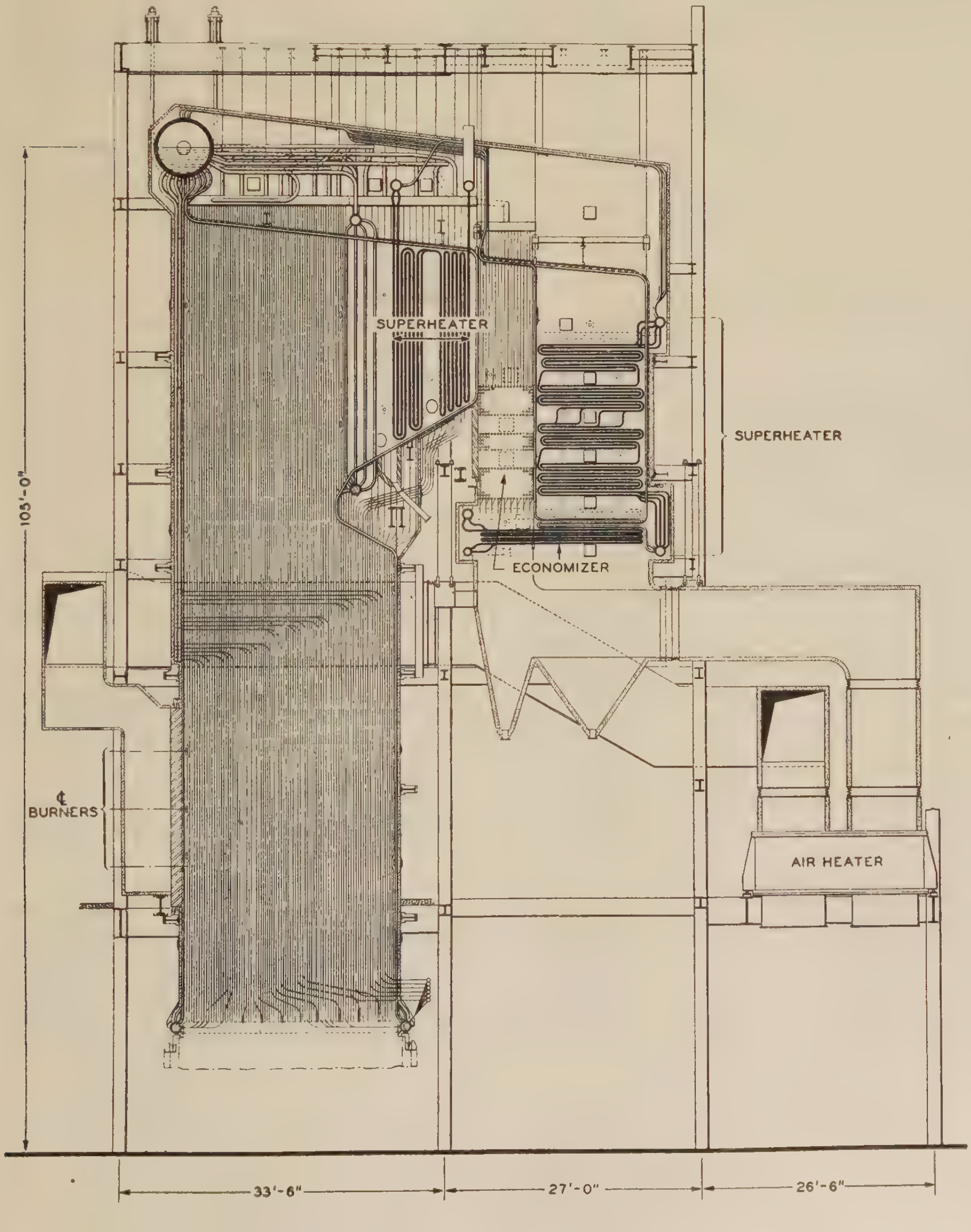


Fig. 10 Boiler for Moss Landing Steam Plant

either coal or oil slag. Experience has indicated that oil-slag pluggage becomes much less troublesome in the lower-temperature zone existing after passing over two banks of superheater tubes, and it is probable that no pluggage would result if the tube spacing were reduced to 1 in. clear between tubes. However, since fuel-oil slag has become progressively more troublesome, it was decided to make the minimum clearance between tubes in the cooler section of the superheater not less than 2-in. The superheater banks were made extremely shallow so that the soot blowers could penetrate the bank properly. Provision for water-washing the superheater, economizer, and air heater was also incorporated.

---

[9] Proposition Division, Babcock and Wilcox Company, New York, N. Y. Mem. ASME.

[10] "History and Performance of the Oil-Refinery Steam-Electric Generating Stations of the Pacific Gas and Electric Company," by V. F. Estcourt, published in this issue of the Transactions, pp. 209–222.

The experience of the Pacific Gas and Electric Company in its own plants and the boiler manufacturer's experience in other plants in California, Florida, and elsewhere, were combined in an effort to design and build a boiler unit that would be as reliable as is humanly possible. If the hopes of the designers and operators are realized in these new units, perhaps the availability records will permit the next expansion program to be of the single-boiler single-turbine design, resulting in a further simplified plant design and operation.

Alf Hansen.[11] This paper describes the power system of Northern California, and gives a clear, factual picture of the background and planning of the current expansion program.

Normal rainfall in California varies from $^1/_2$ in. per year in the desert area in Southern California to 114 in. per year in the northwest area. Water available for hydro-power generation in a dry year is about 40 per cent of a normal year. This can be increased to 60 per cent by the use of storage. Therefore there must be available enough steam-generating capacity to firm up 40 per cent of the hydro capacity.

As the power system grows, available hydro sites become less attractive from a firm power standpoint, a larger percentage of the capacity must be firmed up with steam, and initial cost of development increases. Only about 20–25 per cent of the consumers' dollar pays for the fuel used in a steam-generating plant. Therefore steam generation will assume an increasingly greater share of the power load.

Before the war the power company estimated the division of generating capacity as 60 per cent hydro, 25 per cent steam, and 15 per cent from other sources.

At present (including current expansion) the ratios are 40 per cent hydro, 50 per cent steam, and 10 per cent from other sources.

In the future, best estimates indicate the ratios should be 20–25 per cent hydro, 70 per cent steam, and 5–10 per cent from other sources.

The peak-load curve of the P.G.& E. system from 1914 to date shows the effects of two world wars, drought periods, depressions, and business booms; but if a smooth line is drawn through this curve it indicates an average annual increase of slightly more than 6 per cent. Inspection of the annual peak kilowatt curve in Fig. 2 of the paper shows no increase in 1938, and about 13 per cent in 1947. Therefore there must be enough spare capacity on the system to take care of such unpredictable increases in load as effected by war and business booms.

At the present rate of normal growth, the Northern California power system needs 200,000 kw of additional firm generating capacity every year, a large portion of which will be steam.

Before the war P.G.& E. had about 100,000 kw capacity in old steam units which were shut down and served as "cold stand-by." It is clearly seen from figures given in the paper that these units had to be brought into service, and had to be operated at high load factor during the war. When they are again returned to serve as cold stand-by by the current expansion program, 28 per cent more electrical energy will be produced from the same quantity of fuel. Such improvement in efficiency tends to hold down the cost of electricity in spite of rising prices on equipment and labor. It also helps conserve our resources.

The authors mention that a power-curtailment program became necessary in the spring and early summer of 1948. During the 2-year period 1947–1948, the system-peak demand increased some 400,000 kw over the 1946 peak. Subnormal rainfall during this same period greatly reduced hydro generation and increased irrigation pumping load. During the actual curtailment, the load on the power system was 28 per cent greater than for the same period the preceding year, and in certain irrigation areas 400–1800 per cent greater than during the same period of a normal year.

By close co-operation of the power companies in the West, and pooling of their resources, plus effective use of daylight-saving time, the utility companies were able to pull through without interruption of service and with only a minor rationing program for a short period. This is an achievement of which they may be justly proud.

When the interchange agreement with Southern California Edison Company was terminated in December, 1947, some 70,000 kw of firm power obtained from this source had to be replaced with new steam-generating facilities. However, the first new 75,000-kva unit at Kern near Bakersfield did not go on the line until May 10, 1948, so it was fortunate that the Southern California Edison Company continued to supply power on a nonfirm basis after the expiration date of the firm power agreement.

The paper clearly brings out that irrigation pumping is an ever-increasing portion of the system load. This accounts for the August peak being equal to the December peak in a normal year, and will even exceed it in a dry year. The load factor for the July-August peak is 73–74 per cent, and for the December peak 62–63 per cent.

Centralized control of steam plants has considerable merit from an economic point of view. Reducing the operating force in a steam plant by one man per shift, effects a net annual saving of $15,000. Such a saving will justify a capital expenditure of about $120,000 for laborsaving features and devices.

We have stated that the peak-load curve shows an average annual increase of a little more than 6 per cent. Some time in the future the shape of this curve must change, or the power company would have to double its capacity every 12 years. Experts claim there is a definite relationship between the amount of rainfall in a certain area and the number of people that area can support. According to this theory, the saturation point for the State of California is between 23 and 32 million people. Therefore one might venture to guess or predict that the peak-load curve will continue at the present slope for another 12 years.

---

[11] Turbine Engineer and Specialist, Pacific District, General Electric Company, San Francisco, Calif. Mem. ASME.

The Design of Sewaren Generating Station and No. 1 Unit at Essex Station, Public Service Electric and Gas Company

By F. P. FAIRCHILD,[1] NEWARK, N. J.

The author outlines the more important considerations leading to the design of the 100,000-kw unit at Essex and the four-unit Sewaren Generating Station. Outstanding features are semioutdoor boilers at latitude 41°; 1050 F steam temperature; a simplified plant layout, including the unit system; central control; shaft-end house generators; and all-electric auxiliary drive. The improvements in calculated plant heat rates are shown for the progressive steps of the program. Some of the more important electrical and architectural features are described.

THIS paper is concerned primarily with the new Sewaren Generating Station, but inasmuch as the Essex 100,000-kw unit was a definite step in the development of many Sewaren innovations and principles, the two may be considered almost as progressive steps in the same program.

Preliminary Studies

The Sewaren site, located on an arm of the ocean between Staten Island and the New Jersey coast in Woodbridge Township, was acquired by the Public Service Electric and Gas Company in 1928. With outlet rights of way it consists of approximately 150 acres, of which 35 acres are used in the present development.

Various studies of the method to be used in developing this site, together with probable capital costs and operating economies, had been made from time to time over a period of years. The essentials of the present layout first were developed in a preliminary way the latter part of 1944, at which time consideration was also being given to the removal of an old 22,500-kw unit at Essex, and the substitution of a 100,000-kw, 3600-rpm, modern unit in its place.

Early in 1945, construction of a simple model of a unit station with semioutdoor boilers was begun. The purpose of this model was to provide better visualization of the unit principle of generating-station design, and looking toward Sewaren in particular, it also incorporated outdoor-boiler-plant design and two-level operation. This preliminary model proved to be of considerable use in discussions with the operating department, executives, and others.

General Description—Essex Project

In March, 1945, the work at Essex Generating Station in Newark, N. J., was authorized. The capacity situation was uncertain at the time, but the installation was justified by economies to be obtained which amounted to an annual saving of about 16 per cent of the estimated cost of the installation. The final completed cost was roughly $9,300,000.

It was decided to go ahead with the Essex installation with one boiler and one turbine, and to adopt all features of a complete unit system, namely, one boiler and one turbine generator with a shaft-end generator to carry all essential auxiliaries, without any cross-connections tying into other units in the station. Central control was also adopted as far as was possible in this installation.

Essex embodied a considerable advance in the design of 3600-rpm turbines and generators. The turbine was the first 3600-rpm machine to use 23-in. buckets. These buckets were put through very thorough tests in a special test wheel before installation on the spindle.

Prior to the Essex installation, the largest 3600-rpm generator in existence was rated 65,000 kw. The Essex main generator was rated 95,000 kw and in addition to being the largest generator at that time, it was also the first to be designed for a hydrogen pressure of 25 psi.

Forced- and induced-draft fans, air preheaters, and flue-dust collectors were placed above the boiler, out of doors. For this outdoor service the manufacturers, in co-operation with Public Service engineers, developed fans with air-cooled bearings, large motors of the totally enclosed fan-cooled type, and air-to-oil coolers for hydraulic couplings, to eliminate water-cooling connections of any kind. In the interest of simplicity and the reduction of make-up, it was decided to omit the usual auxiliary steam header. This departure involved the development of a motor-driven condenser air removal pump, a satisfactory substitute for the usual steam-driven auxiliary oil pump of the turbine, the use of air soot blowing, and the elimination of the reserve steam-driven boiler feed pump. Commerical operation of the Essex unit started in December, 1947.

General Description—Sewaren Project

On January 21, 1946, authorization was given for the construction of a generating station at Sewaren, consisting of two 100,000-kw units to operate at 1500 psi and 1050 F. These units were required to meet the system load as forecast for 1948. Substantial operating savings were indicated. These two units started commercial operation at the end of November, 1948. A time-table of important dates concerning these two units appears in Table 1.

A third Sewaren unit was authorized January 9, 1947, because of the rapid growth of load on the system. This unit is scheduled to operate in the fall of 1949. It is a twin of the second unit.

On December 11, 1947, funds were appropriated for the purchase of the fourth turbine generator with step-up transformers at Sewaren, in order to obtain shipment early in 1951. No. 4 is similar to the first three units except that it will be rated at 125,000 kw, and will operate with reheat to 1000 F. Prior to this time a 100,000-kw nominal rated turbine generator was the largest 3600-rpm single-shaft machine available. Studies indicated that the larger equipment can be installed in the space provided for the fourth unit without crowding, and that substantial fuel savings will result from the improved efficiency of the reheat cycle.

---

[1] Chief Engineer, Electric Engineering Department, Public Service Electric and Gas Company. Mem. ASME.

Contributed by the Power Division and presented at the Semi-Annual Meeting, San Francisco, Calif., June 27–30, 1949, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 49—SA-15.

TABLE 1 TIME-TABLE TO DESIGN, BUILD, AND ERECT FIRST TWO 100,000-KW UNITS AT SEWAREN GENERATING STATION

	Starting dates
Design	Jan. 1946
Purchases:	
Turbine generators	March 1946
Boilers	May 1946
Erection in field:	
Excavation	Jan. 1947
Pile driving	March 1947
Concrete	May 1947
Structural steel	July 1947
Boiler:	
No. 1	Oct. 1947
No. 2	Nov. 1947
Turbine generator:	
No. 1	May 1948
No. 2	June 1948
Commercial operation:	
Unit No. 1	Dec. 1, 1948
Unit No. 2	Nov. 30, 1948

The present estimated unit cost of the four-unit station is $132 per kw for name-plate conditions. This includes all electrical equipment to the high-voltage side of the transformers.

The approach to the basic design of Sewaren was the same as Essex, namely, to develop the unit system to the greatest extent practicable. Here, however, as architectural and structural limitations did not exist, it was possible to accomplish much more in this direction. The Sewaren design makes use of all the features developed for Essex plus two-level operation, semioutdoor boilers, and central control of units in pairs.

Sewaren Plant Arrangement Compared to Essex

The Essex unit was installed in the area occupied for 32 years by a 22,500-kw turbine generator and four boilers. Fig. 1 is a view of the Essex turbine room with the new unit in the foreground. No. 2 unit, next to it, is a twin of the original No. 1. The new equipment was arranged to suit the building; whereas at Sewaren, the building was designed to accommodate the equipment. The cross sections of the two plants show this difference. At Essex, Fig. 3, on account of limited floor area, the fans, air preheaters, flue-dust collector, and stack are above the boiler; coal is brought from the existing bunker by conveyer to a hopper which feeds the pulverizers.

In contrast to this, the Sewaren auxiliary equipment is laid out as near ground level as possible, Fig. 2. The circulating-water pumps are at the screen well; the fans, air preheaters, and dust collectors are out of doors; the stacks are on separate foundations; the pulverizers are placed under the coal bunker. The plant is spread out horizontally.

Semioutdoor Boilers

The Sewaren boilers are of the semioutdoor type in which the building encloses the front and one side of each boiler. The exterior of the boilerhouse is shown in Fig. 4. The form of the building in the vicinity of the boilers is shown more clearly in the simplified sketch, Fig. 5. This type of construction saved about $1,000,000 in first cost of three units. Because of the smaller building volume and reduced heat radiation in the boilerhouse, the building is easier to ventilate and keep clean.

Boiler overhauls will be on an annual basis and will be scheduled for warm weather. By far the greater portion of the boiler-maintenance work is performed inside the boiler furnace, gas passes, and boiler drums. Arrangements have been made to keep the inside of the furnaces and gas passages warm during winter shutdowns, not only for maintenance work but also to prevent freezing of the superheaters.

Boiler Design

The designs of the Essex and Sewaren steam-generating equipment are particularly adapted to each plant layout. At Essex, where the limitation was floor area, the steam generator and its auxiliaries occupy an area of about 70 ft × 60 ft. Coal hopper, pulverizers, coal piping, and burners are fitted compactly under the superheater, economizer, and uptake duct. Coal is brought from the station bunker to the hopper by conveyers.

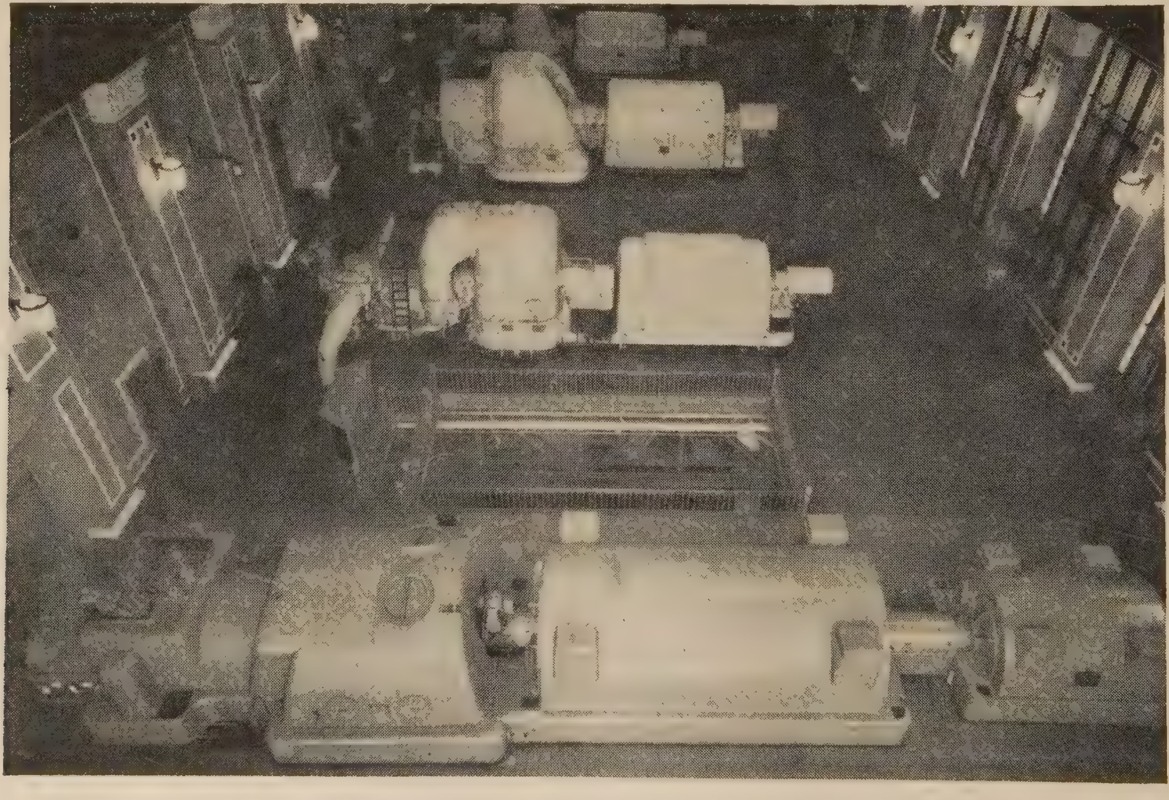


Fig. 1 Turbine Generator of 100,000 Kw Installed at Essex in Area Formerly Occupied by 22,500-Kw Twin of Adjacent Unit

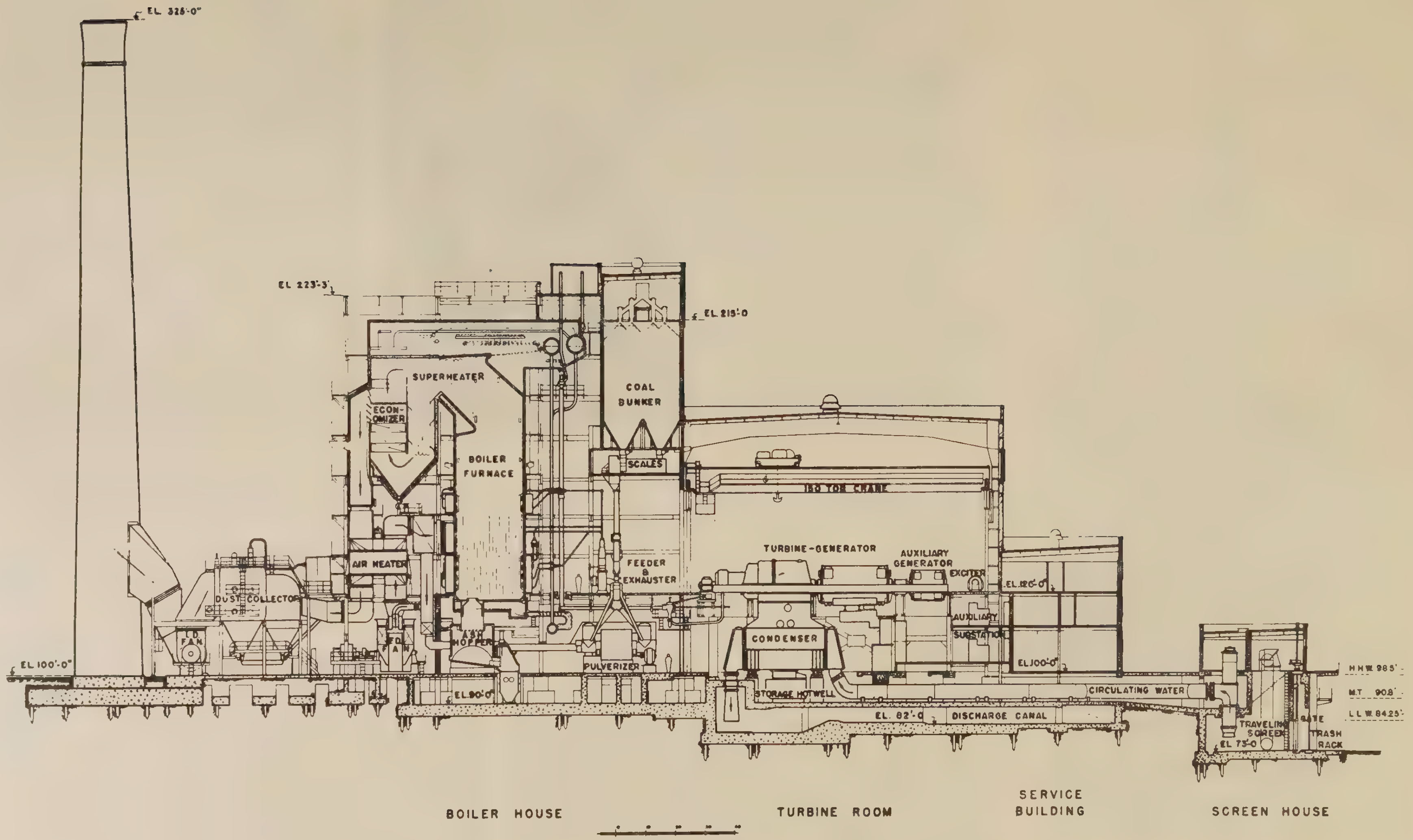


Fig. 2 Plant Cross Section, No. 2 Unit—Sewaren

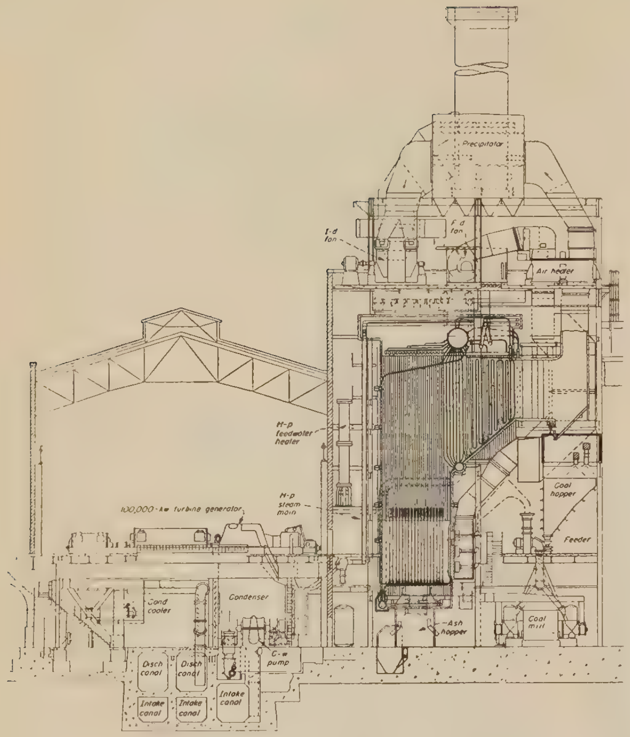


Fig. 3 (*left*) Plant Cross Section, No. 1 Unit—Essex

The Essex boiler, Fig. 6, is a two-drum radiant type, having a continuous rating of 900,000 lb per hr and a maximum rating of 1,000,000 lb per hr for 4 hr with throttle conditions of 1250 psi and 1000 F. The slag-tap furnace, which has a volume of 47,800 cu ft, is divided into two equal volumes by a center waterwall having interconnecting ports in the flame area. Three horizontal ball-type pulverizing mills serve the twelve combination pulverized coal-mechanical atomizing oil burners. Outlet steam temperature is controlled by means of a spray-type attemperator placed between the two sections of the superheater. Auxiliaries include two forced- and two induced-draft fans, two Ljungström air heaters, an economizer, and an electrostatic dust collector.

At Sewaren, where the design objectives included the semioutdoor feature, two-level operation, and the utmost simplicity, the area occupied by the steam generator and auxiliaries is roughly 160 ft × 50 ft, about twice the area used at Essex. The drums and superheater outlet header are offset from the furnace so that they are housed in the building. This facilitated the semioutdoor design of the boilerhouse.

Each of the first three Sewaren boilers, Fig. 7, is of the three-drum, radiant type, having a continuous rating of 850,000 lb per hr, and a maximum of 950,000 lb per hr for 4 hr with throttle conditions of 1500 psi and 1050 F. The slag-tap furnace, which has a volume of 52,450 cu ft, is arranged for tangential corner firing at two levels, with combination pulverized-coal and oil burners vertically adjustable to permit positioning of the turbulent burning zone in order to obtain optimum slagging conditions on the walls and floor of the furnace and to effect a certain amount of superheat control. Final steam temperature is controlled by a damper which by-passes part of the furnace gases from the primary superheater to the economizer. As at Essex, there are three pulverizers, two Ljungström air heaters, an economizer, and two forced- and two induced-draft fans. The dust collector is a combination mechanical-electrostatic unit, with the mechanical section ahead of the electrostatic.

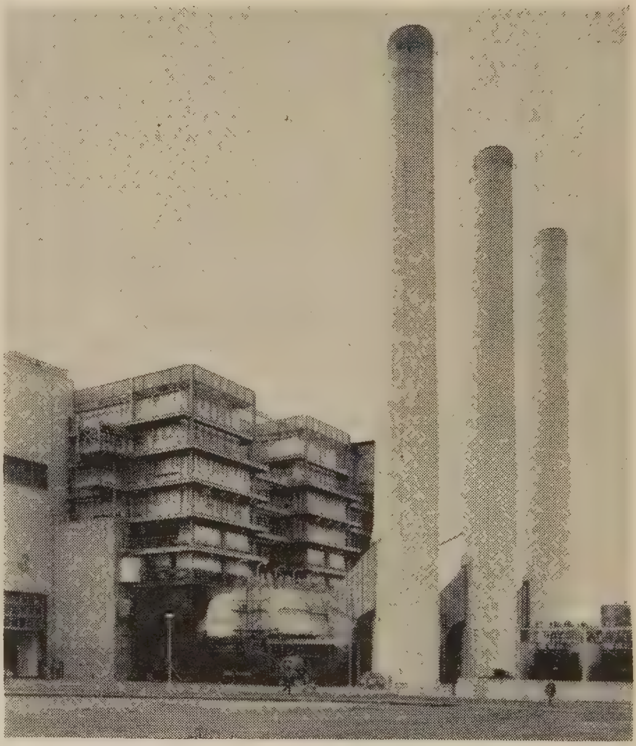


Fig. 4 Semioutdoor Boilers—Sewaren

Turbine Design and Heat Rates

At its best point, the Essex unit has a calculated plant heat rate (based upon turbine guarantees) of 10,400 Btu per net kwhr. Steam conditions are 1250 psi, 1000 F, and the turbine has a double-flow low-pressure cylinder, Fig. 8. This unit operated 8149 hr of the 8784 hr in 1948 at an average turbine load of about 97,800 kw. The capacity factor was 90.7 per cent. The average heat rate for the year was 10,500 Btu per kwhr, which is approximately the calculated rate for that load. The average circulating-water temperature was 60.6 F; average back pressure was 1.36 in. Hg. The unit operates regularly at a rating of 110,000 kw.

No. 1 unit at Sewaren is essentially similar to Essex No. 1, except that the steam conditions are increased to 1500 psi and 1050 F. The improvement in heat rate is 4.6 per cent at the best point. This increment includes a one-point improvement

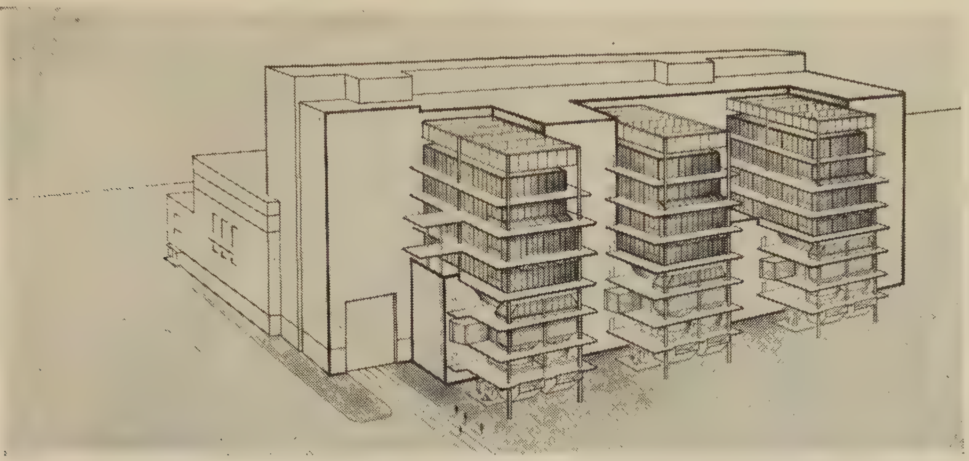


Fig. 5 Sketch of Exposed Parts of Boilers—Sewaren

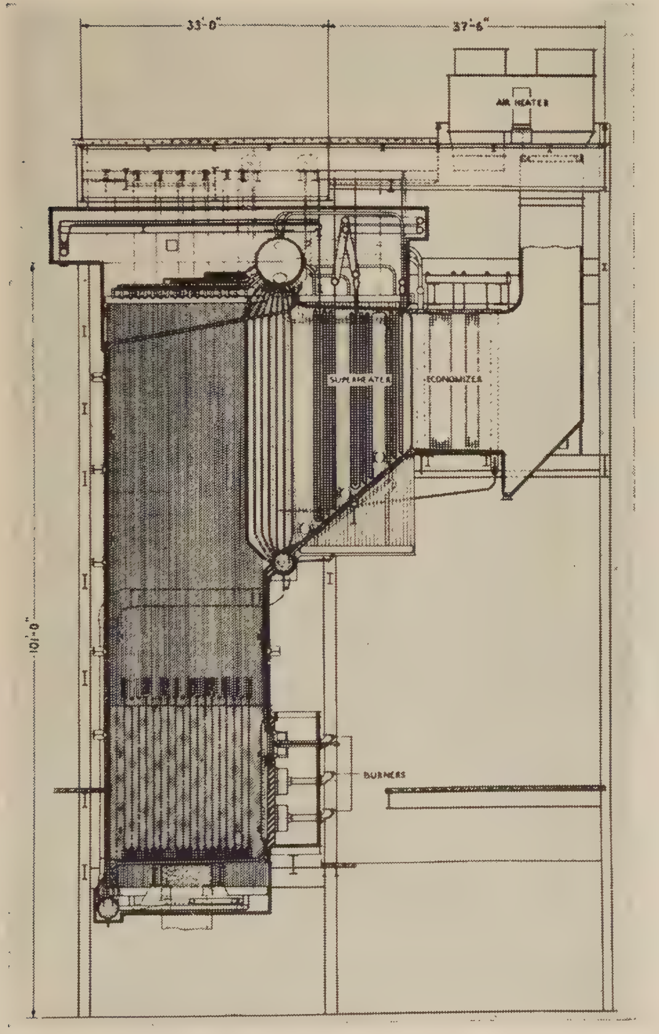


Fig. 6 Boiler Cross Section—Essex

in boiler efficiency. The cross section of the turbine is shown in Fig. 9.

No. 2 and 3 units at Sewaren are similar to No. 1 unit except that they have triple-flow low-pressure cylinders. The improvement over No. 1 unit (Sewaren) is 1.4 per cent at the best point. The cross section of the turbine is shown in Fig. 10.

No. 4 unit is similar to No. 2 and 3 at Sewaren except that it is a larger machine, and the steam is reheated to 1000 F. The improvement over No. 2 and 3 units is 4 per cent at the best point, Fig. 11.

The calculated plant heat rates for the units are shown in Fig. 12. The heat-balance diagram for No. 2 and 3 units at Sewaren, Fig. 13, is typical. The system operates without a heater-drain pump. Although the diagram does not show it, the hot-well pump is of the re-entry type. The condensate re-entry is made after passing through the condensate cooler, generator, hydrogen and air coolers, and oil and miscellaneous coolers.

Central Control

Unit design, with only one boiler per turbine, greatly facilitates central control in a modern electric generating station. For example, the control-panel areas of No. 1 unit at Sewaren compared with the corresponding areas of No. 6 unit at Burlington, a 125,000-kw cross-compound unit with two cross-connected boilers installed in 1943, show some interesting figures. At Sewaren there are 521 sq ft of panel in one location; at Burlington there are 695 sq ft in five separate locations. This concentration of control results in better co-ordinated operation and a reduction in operating attendance.

The arrangement of the control panel at Essex is shown in Fig. 14. Care was taken to make it short so that the operator can see the maximum number of instruments and can reach the greatest number of controls with the least movement. Instruments and controls of lesser importance were placed on the end of the panel wings to make room for the more essential items on the front and center of the panel. While there is no distinct separation between boiler, turbine, and auxiliary boards, the controls are placed on the panels in relative arrangement similar to that of the equipment controlled. The board is arranged so that, in the future, it can be enclosed in a separate room. The plan of the Essex operating floor, Fig. 15, shows the location of the control board. In this installation the generator and electrical auxiliary supply controls were located in an existing electrical control room.

The two Sewaren control rooms are tee-shaped, with the electrical benchboard, Fig. 16, forming the crossbar. The operator sees the two turbine generators through windows just above the benchboard. The other sides of the room, Fig. 17, are occupied by the panels controlling the turbines, boilers, and their auxiliaries. At the rear, there are two windows looking toward the boilers. There are three operators in the control room. Normally, one operates the electrical benchboard of two

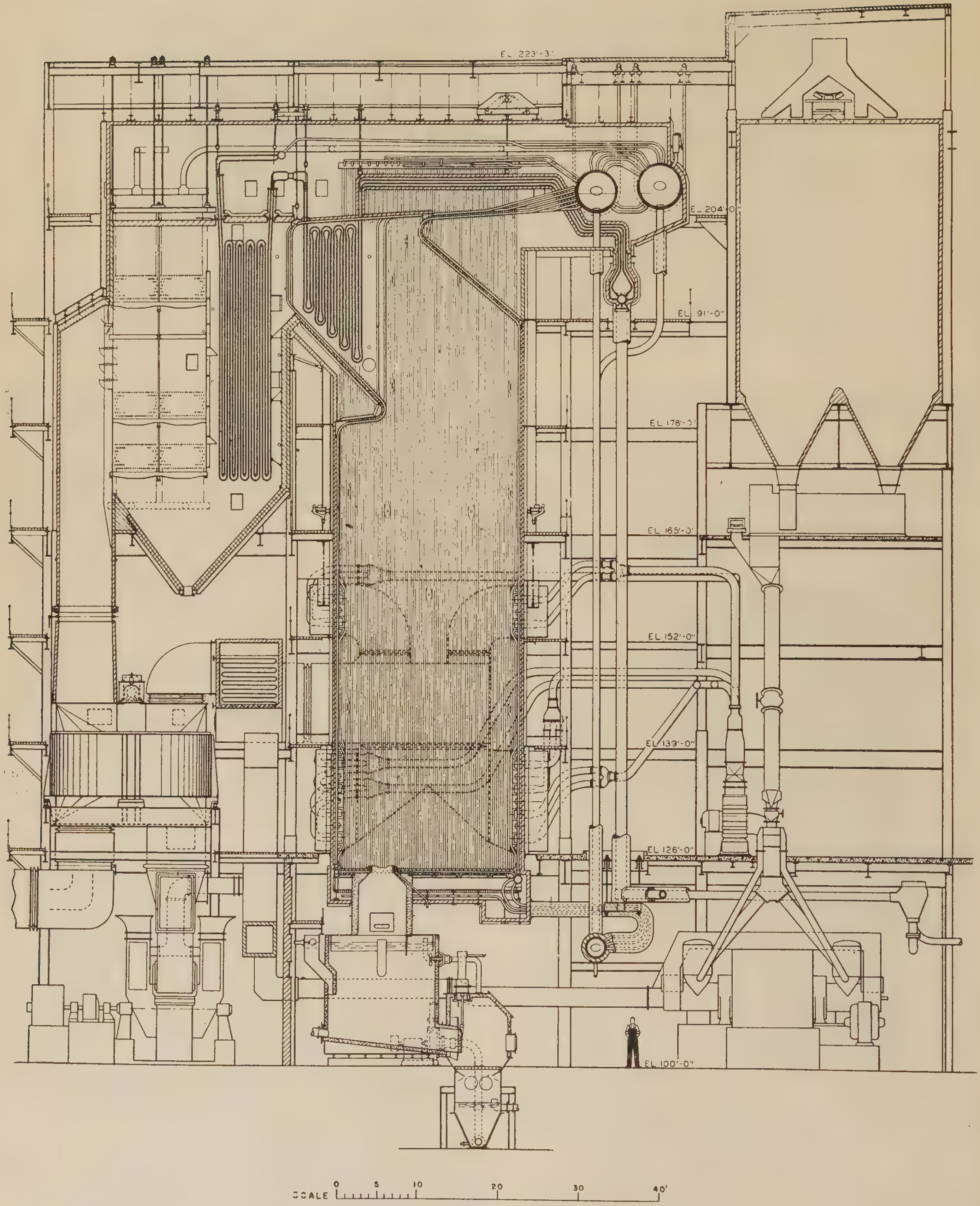


Fig. 7 Cross Section, No. 1, 2, and 3 Boilers—Sewaren

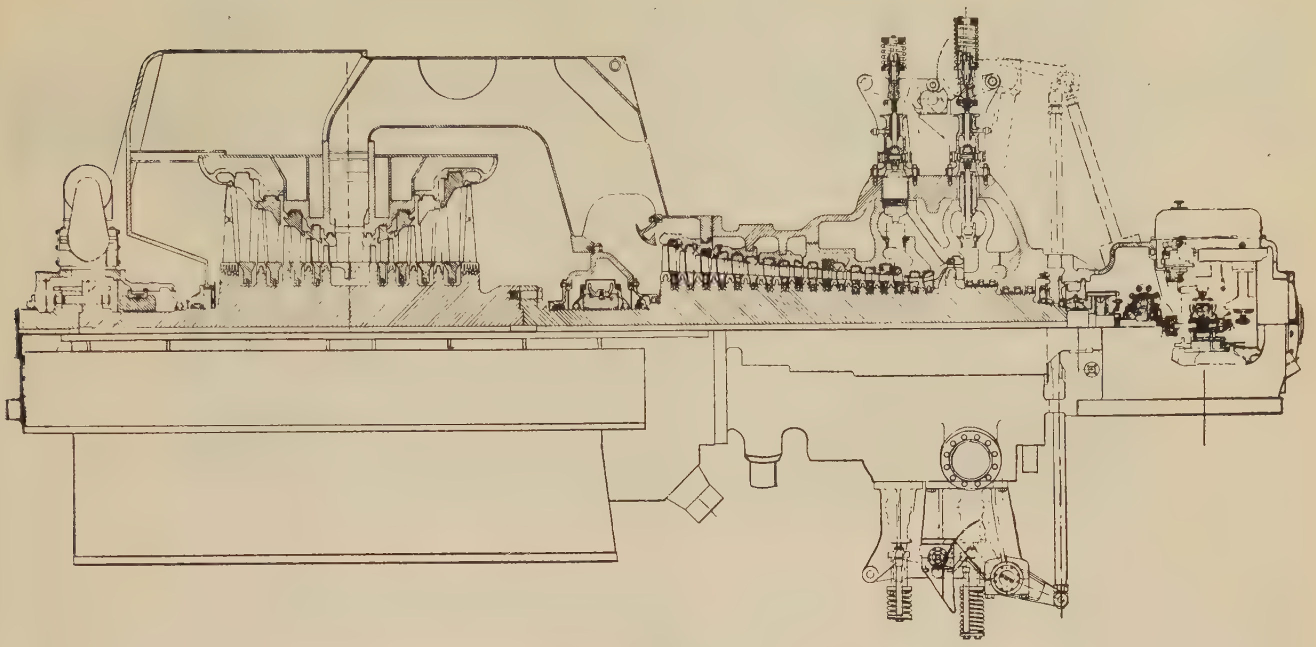


Fig. 8 Turbine Cross Section—Essex

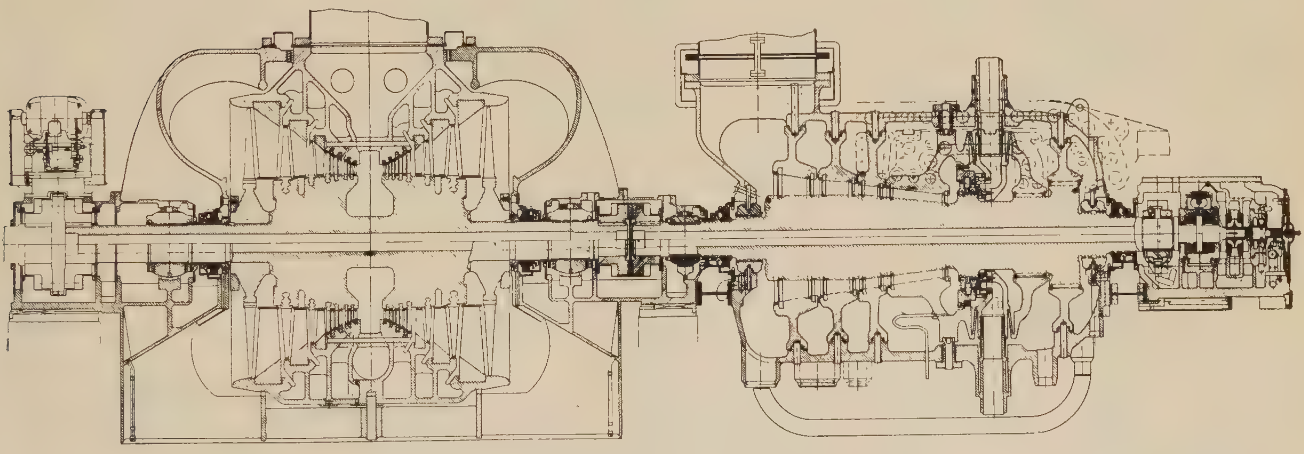


Fig. 9 Cross Section, No. 1 Turbine—Sewaren

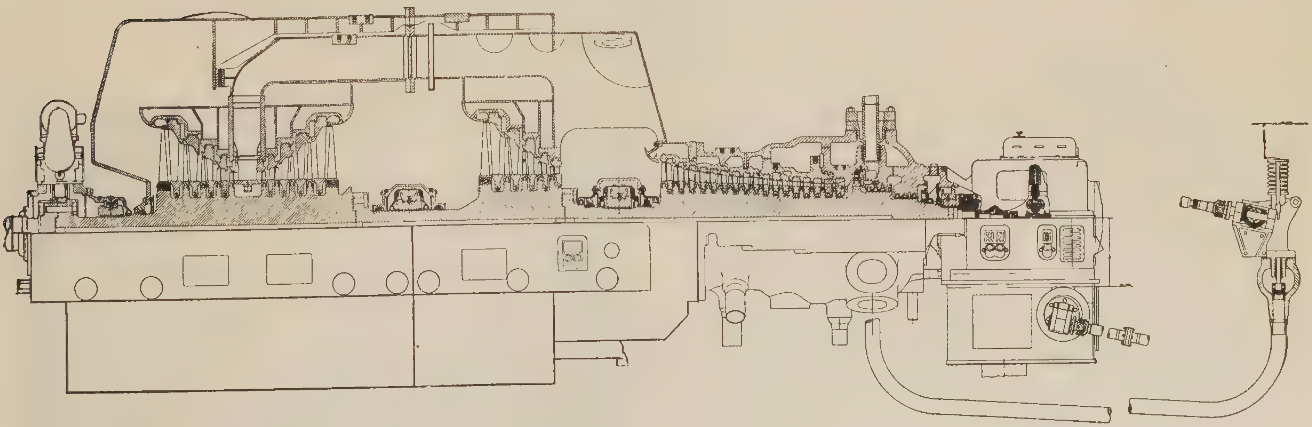


Fig. 10 Cross Section, No. 2 and 3 Turbines—Sewaren

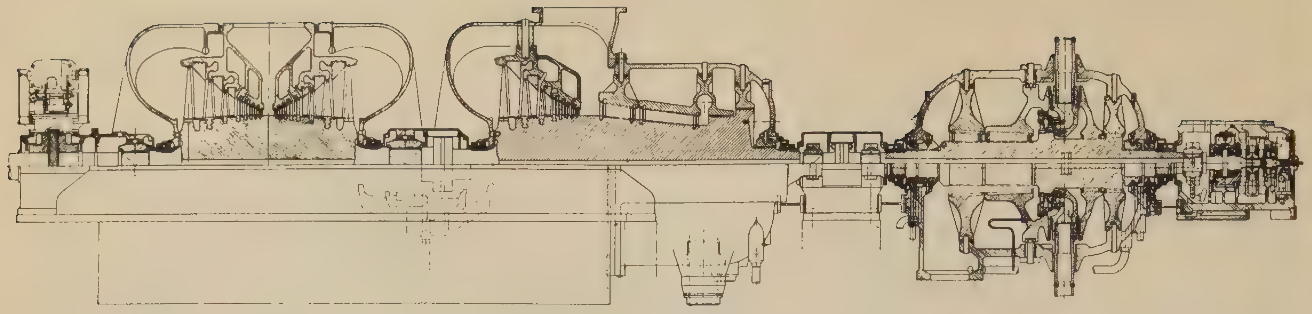


Fig. 11 Cross Section, No. 4 Turbine—Sewaren

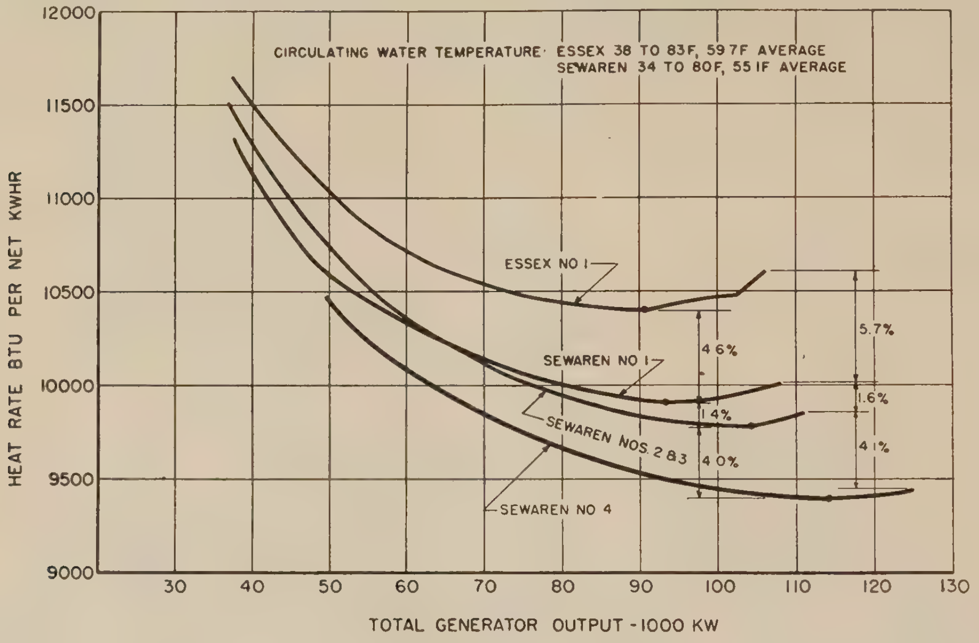


Fig. 12 Calculated Plant Heat Rates

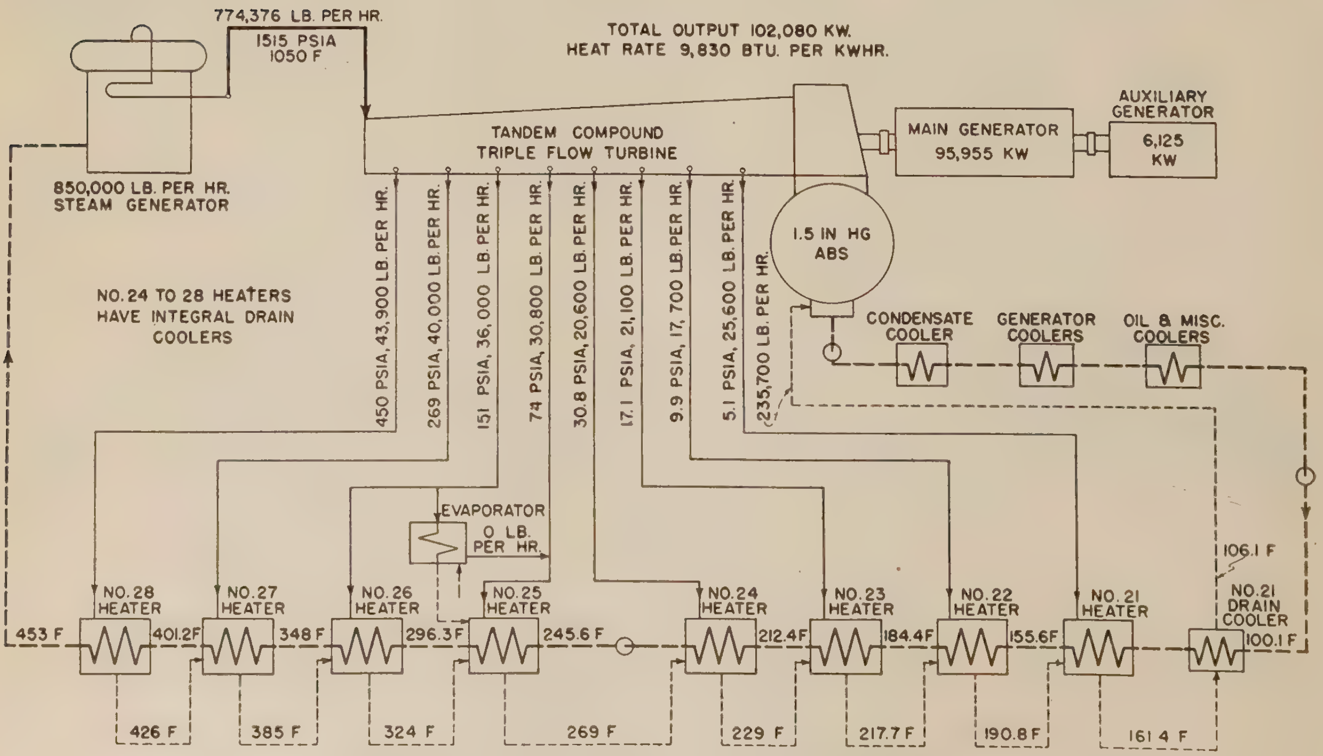


Fig. 13 Heat-Balance Diagram, No. 2 and 3 Units—Sewaren

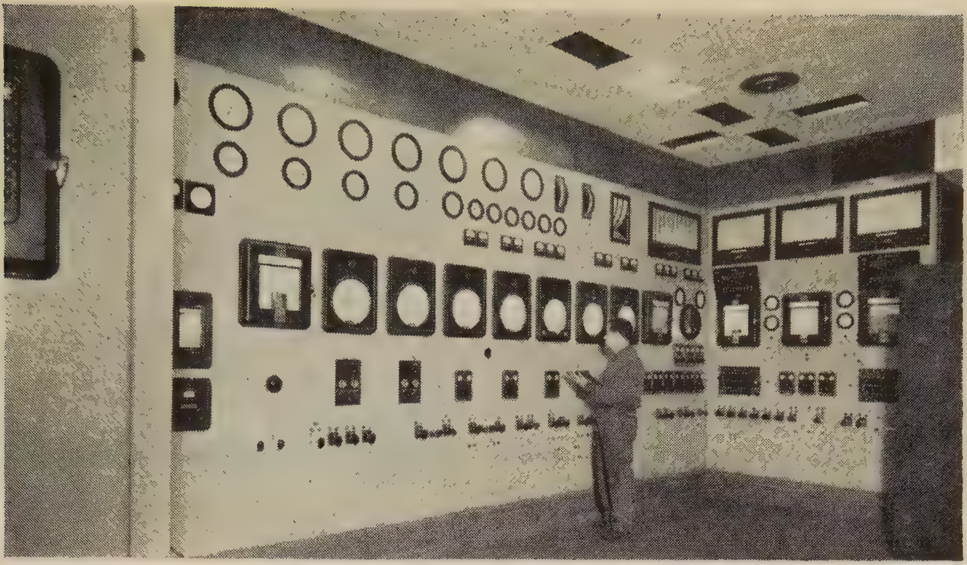


Fig. 14 Essex Control Panel

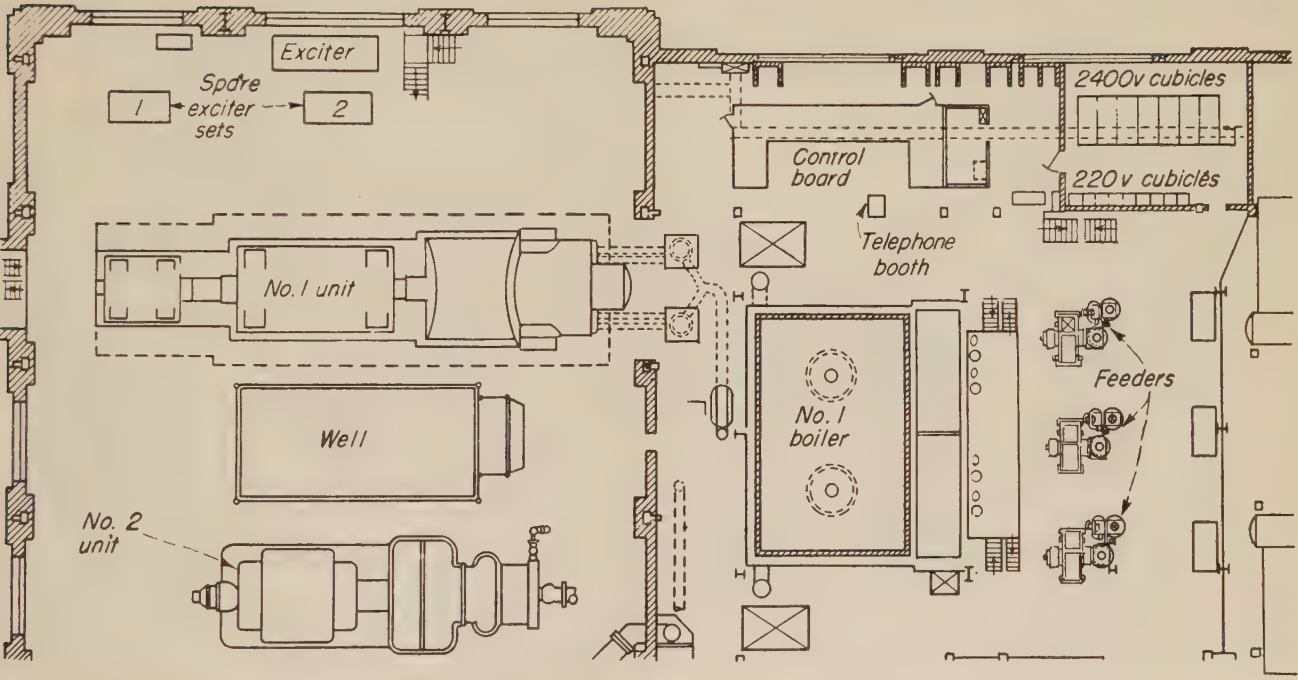


Fig. 15 Plan of Main Operating Floor—Essex

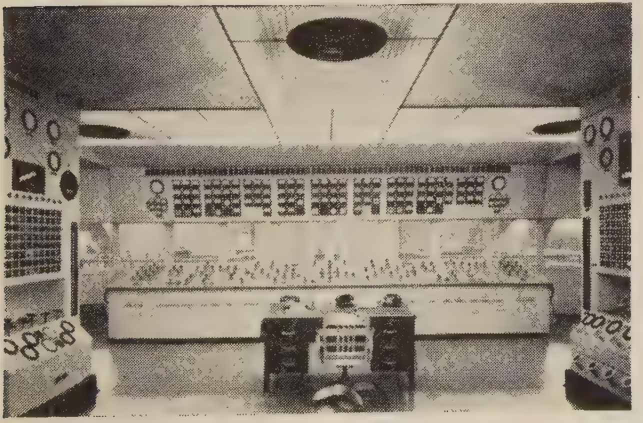


Fig. 16 Electrical Section of Sewaren Control Room

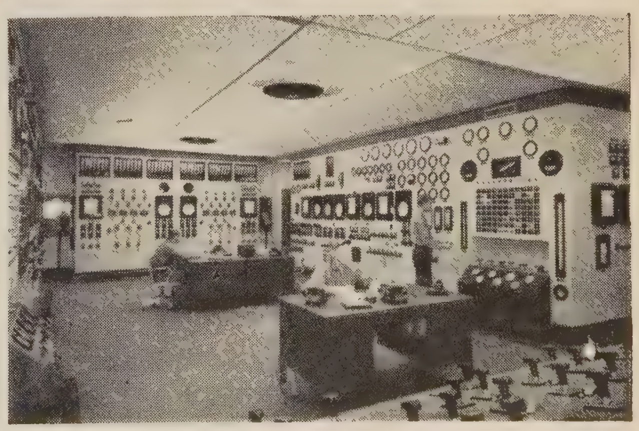


Fig. 17 Mechanical Section of Sewaren Control Room

units, and each of the other two operates the mechanical equipment of one unit. However, the operators are trained to handle both mechanical and electrical controls, and, at present, each one rotates around the three operating jobs in the control room. The control rooms are air-conditioned and soundproofed. Special attention was given to the lighting, particularly of the vertical panels, for which totally indirect lighting is provided by means of a cove and arched-ceiling design. In the areas between sections of arched ceiling, vinylite eggcrate plastic was used to provide a luminous ceiling, which avoids the contrast between lighted and unlighted areas which a solid ceiling would have given. The final design was the result of studies made of a mock-up, erected in the basement of the Public Service Terminal Building in Newark, N. J.

Each control room is situated at the center of the area occupied by the main equipment under its supervision, Figs. 18 and 19.

High-Temperature Parts

The steels used in the pressure parts exposed to the highest-temperature steam are listed in Table 2.

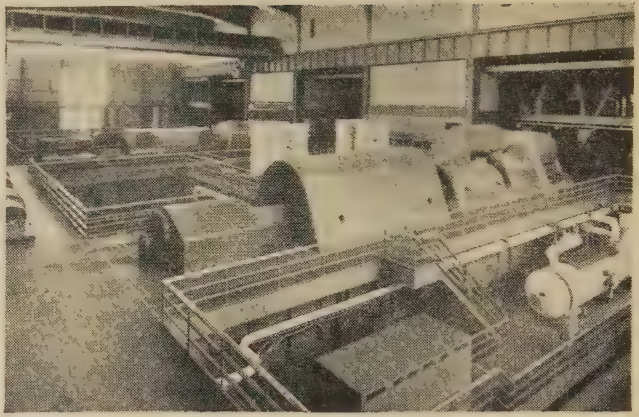


Fig. 19 No. 1 and 2 Turbine Generators and Control Room—Sewaren

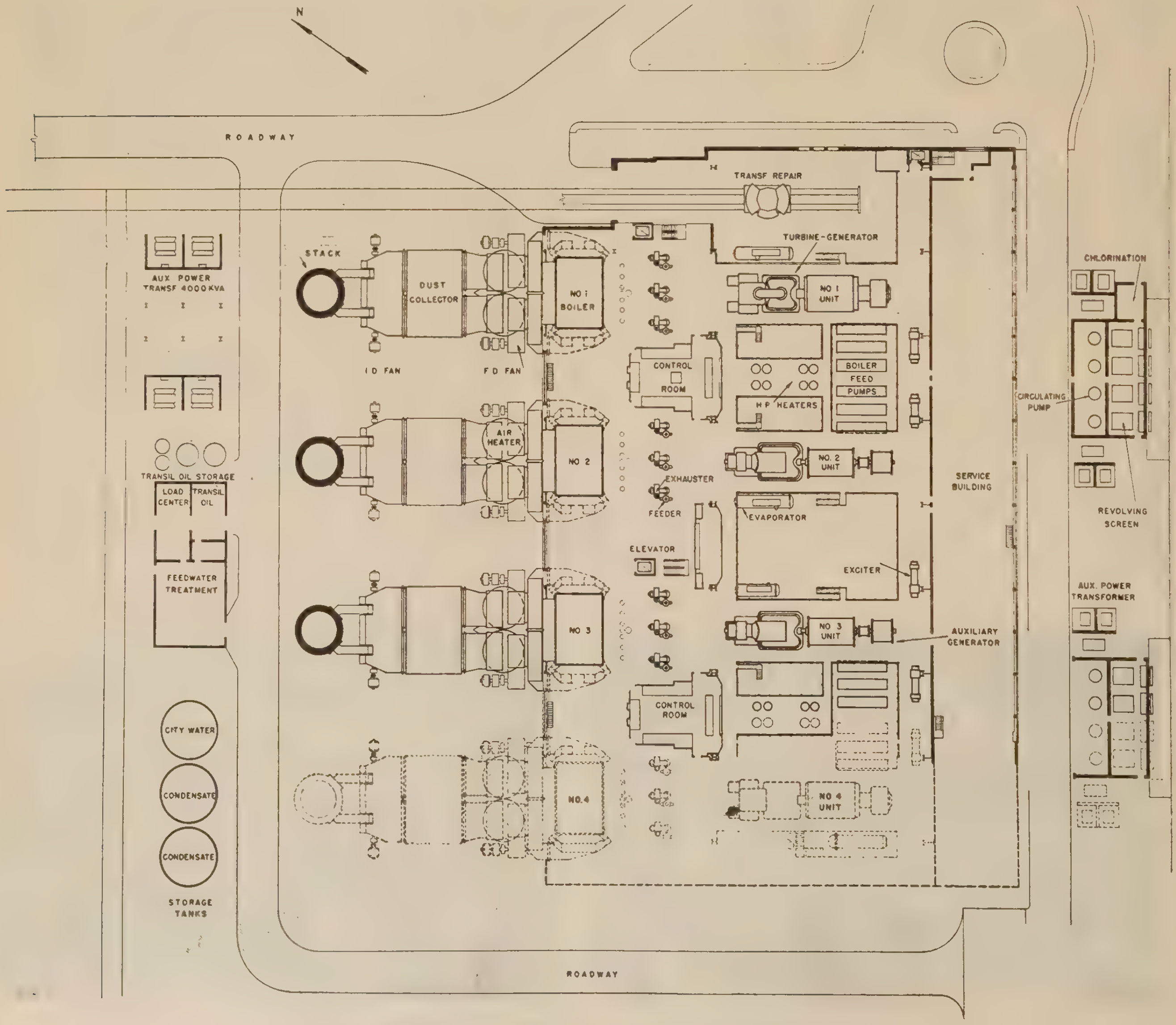


Fig. 18 Plan of Upper Operating Floor—Sewaren

TABLE 2 STEELS USED IN PRESSURE PARTS EXPOSED TO HIGHEST-TEMPERATURE STEAM

	Essex	Sewaren	
	No. 1	No. 1	No. 2 and 3
Steam temperature	1000 F	1050 F	1050 F
Superheater tubes	9 Chromium 1.3 Molybdenum	19 Chromium 11 Nickel Titanium stabilized	19 Chromium 11 Nickel Titanium stabilized
Steam piping	$2\frac{1}{4}$ Chromium 1 Molybdenum	3 Chromium 1 Molybdenum	3 Chromium 1 Molybdenum
Turbine valves	0.14 Vanadium 1.1 Molybdenum	18 Chromium 8 Nickel Columbium stabilized	17 Chromium 14 Nickel 2 Molybdenum Columbium stabilized
Turbine cylinder	0.14 Vanadium 1.1 Molybdenum	$2\frac{1}{4}$ Chromium 1 Molybdenum	17 Chromium 14 Nickel 2 Molybdenum Columbium-stabilized
Turbine buckets	12 Chromium	13 Chromium	12 Chromium 5 Cobalt 3 Tungsten 0.15 Molybdenum

NOTE: All compositions given in per cent of alloying materials.

At Sewaren, each unit has two welded joints between the austenitic steel of the stop valves and the ferritic steel of the steam piping. As the stainless steel expands about 50 per cent more than the chrome-moly steel in the range from 70 to 1050 F, there was some concern about the adequacy of a welded joint between the two metals in heavy wall thickness. With the co-operation of the boiler, turbine, and pipe suppliers, a cyclic-heating test was made on several full-size, heavy-wall, welded pipe joints between austenitic and ferritic steels. This test has been des-

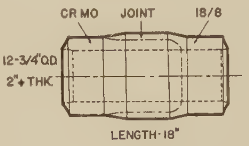


FIG. 20 KELCALOY TRANSITION PIECE

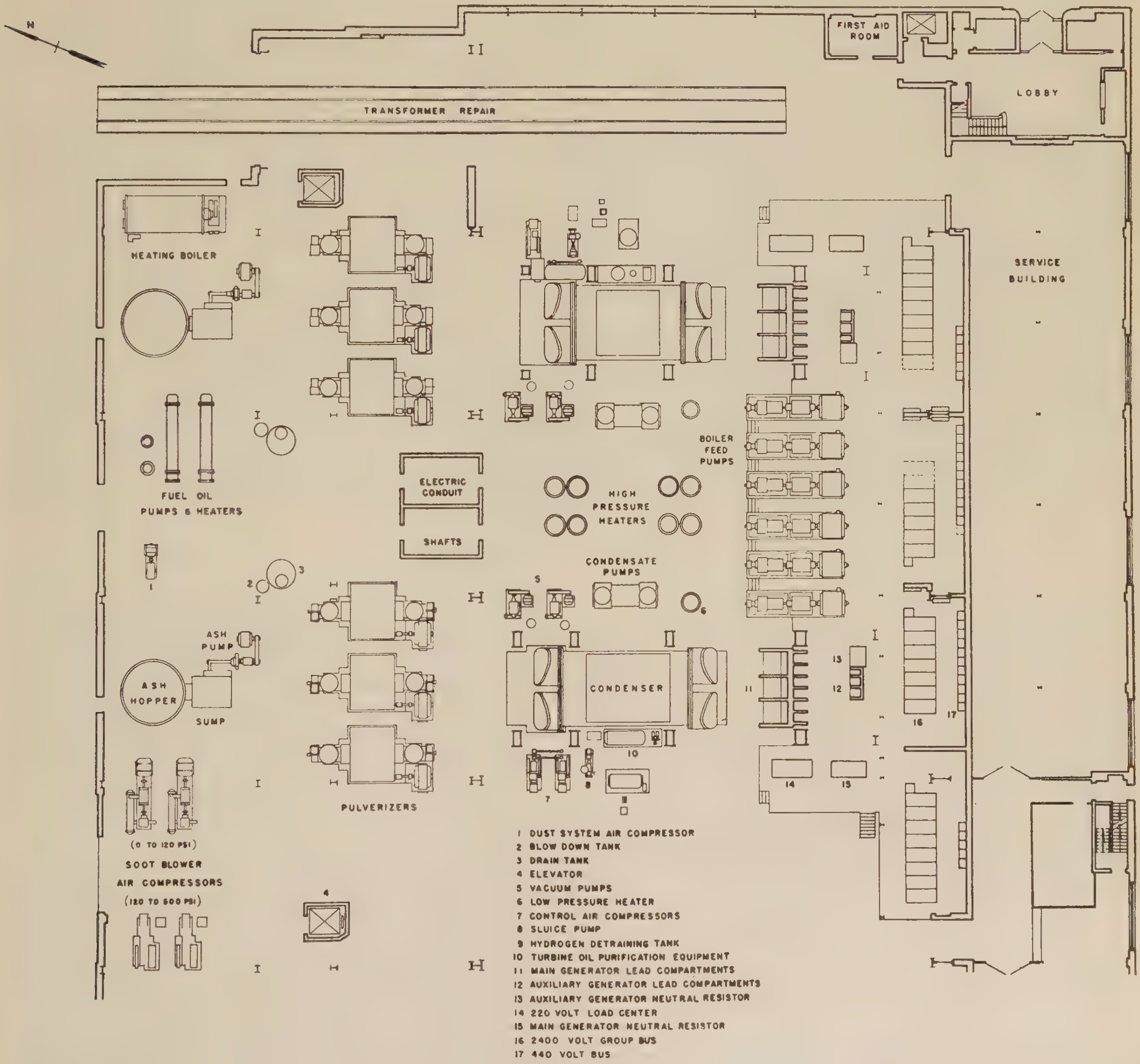


FIG. 21 PLAN OF LOWER OPERATING FLOOR—SEWAREN

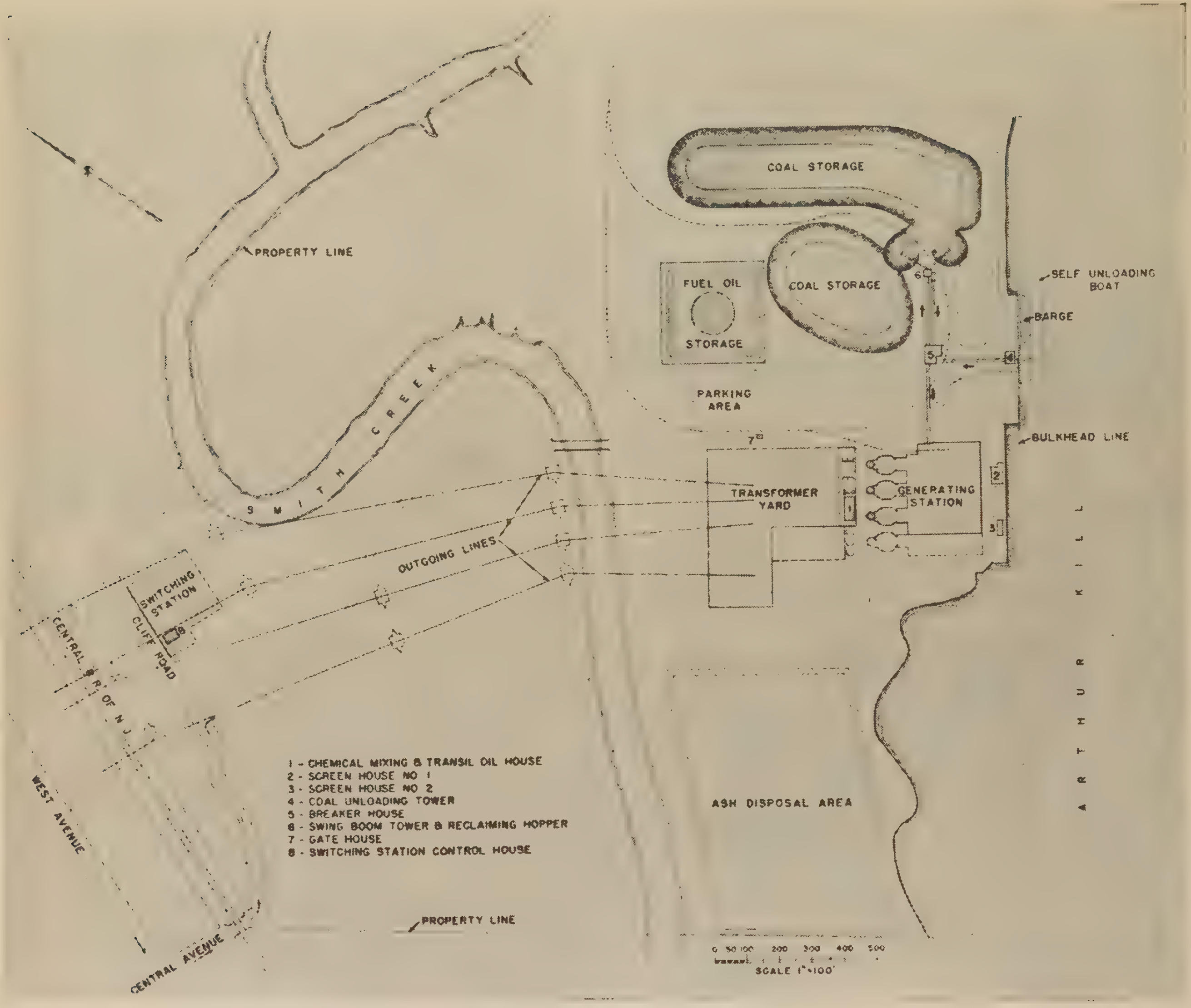


Fig. 22 Sewaren Property Map

cribed fully by H. Weisberg.[2] The Kelcaloy transition piece finally selected is shown in Fig. 20. The purpose of this arrangement is that the major portion of the interface between the dissimilar metals is longitudinal to the pipe rather than transverse, and therefore is subject principally to shear stress resulting from internal pressure or bending. If failure occurs at the joint between the two metals it should appear as a lap or lamination rather than as a separation straight across the pipe. The transition piece is welded to the stop valves in the shop, leaving a ferritic weld to be made in the field.

The spindles of the stop and control valves of No. 2 and 3 turbines at Sewaren are cooled with steam from the intermediate header of the superheater. No. 1 turbine valves are not provided with any cooling arrangement.

Auxiliary Power

Shaft-end auxiliary generators are used to provide the essential auxiliaries with a direct source of power free from outside disturbances. Normally, each auxiliary generator carries the essential auxiliaries of its own unit. It is rated 7500 kw at 2400 volts, 80 per cent power factor, and is separately excited. The 2400-volt auxiliary buses of each unit can be fed either from the auxiliary generator or from auxiliary transformers connected to an outside source. The two are never paralleled. Relays are installed to throw over automatically from generator to transformer in case of loss of voltage on the generator. Manual throw-over in the other direction is provided for transferring auxiliaries from transformer to auxiliary generator when starting up.

Auxiliaries

Steam from the high-pressure boilers is used only to run the main turbines. All auxiliaries are electric-motor-driven. At Sewaren, steam for heating fuel oil, mixing chemicals, and heating the building is supplied normally from the evaporators. There is a heating boiler for use in the event of a complete station shutdown. Every effort has been made to reduce make-up to a minimum.

With the shaft-end auxiliary generator as the source of auxiliary power, the elimination of steam-driven auxiliaries becomes a relatively easier step in station design. One of the problems in

---

[2] "Cyclic Heating Test of Main Steam Piping Joints Between Ferritic and Austenitic Steels, Sewaren Generating Station," by H. Weisberg, Trans. ASME, vol. 71, 1949, pp. 643–664.

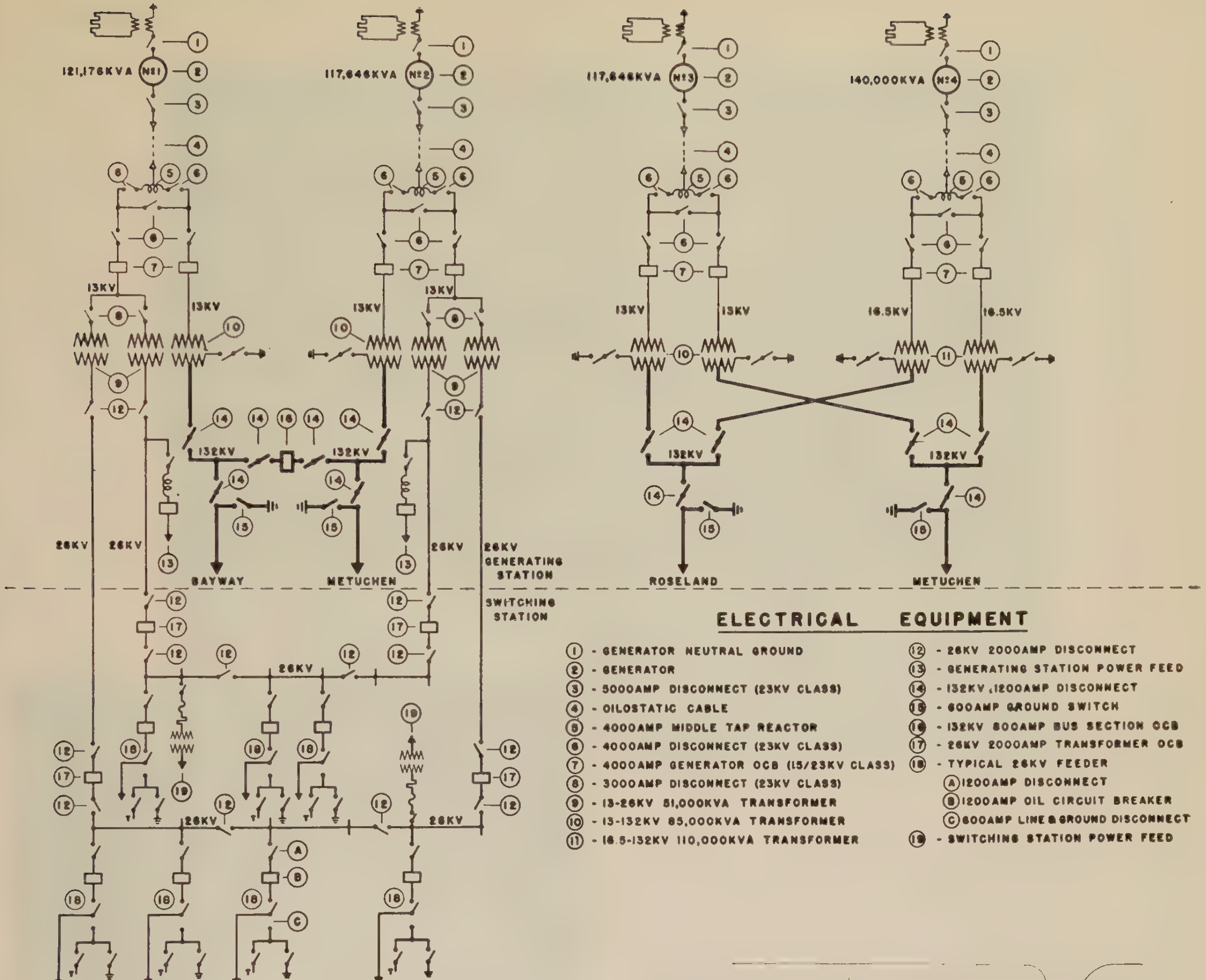

121,176KVA
№1
117,646KVA
№2
117,646KVA
№3
140,000KVA
№4
13KV
16.5KV
132KV
26KV
BAYWAY
METUCHEN
ROSELAND
METUCHEN
26KV GENERATING STATION
SWITCHING STATION
ELECTRICAL EQUIPMENT
① - GENERATOR NEUTRAL GROUND
② - GENERATOR
③ - 5000AMP DISCONNECT (23KV CLASS)
④ - OILOSTATIC CABLE
⑤ - 4000AMP MIDDLE TAP REACTOR
⑥ - 4000AMP DISCONNECT (23KV CLASS)
⑦ - 4000AMP GENERATOR OCB (15/23KV CLASS)
⑧ - 3000AMP DISCONNECT (23KV CLASS)
⑨ - 13-26KV 51,000KVA TRANSFORMER
⑩ - 13-132KV 85,000KVA TRANSFORMER
⑪ - 16.5-132KV 110,000KVA TRANSFORMER
⑫ - 26KV 2000AMP DISCONNECT
⑬ - GENERATING STATION POWER FEED
⑭ - 132KV 1200AMP DISCONNECT
⑮ - 600AMP GROUND SWITCH
⑯ - 132KV 800AMP BUS SECTION OCB
⑰ - 26KV 2000AMP TRANSFORMER OCB
⑱ - TYPICAL 26KV FEEDER
Ⓐ 1200AMP DISCONNECT
Ⓑ 1200AMP OIL CIRCUIT BREAKER
Ⓒ 600AMP LINE & GROUND DISCONNECT
⑲ - SWITCHING STATION POWER FEED

FIG. 23 ULTIMATE CONNECTION DIAGRAM OF THE FOUR SEWAREN GENERATORS

this connection was the substitution of motor-driven, positive-displacement, rotary vacuum pumps for the usual steam-jet ejectors for the removal of noncondensable gases from the condenser.

Before proceeding with this new departure, a Kinney pump of this type was tried out on No. 10 condenser at Marion Generating Station for about 1000 hr. This test revealed that most of the troubles were related to separating the lubricating oil from the water vapor associated with the gases coming from the condenser air coolers. As these difficulties appeared to be surmountable, it was decided to use Kinney pumps at Essex because they offered the following advantages over the steam-jet air pumps: (*a*) Sufficient capacity at low vacuum to eliminate the need of starting evactors; (*b*) adaptability to remote push-button starting and stopping; and (*c*) more complete removal of corrosive gases such as ammonia and carbon dioxide from the system, as there are no inter- and after-condensers (or, alternatively, no wastage of condensate to get rid of these gases).

It was evident after a few months' operation of the Essex pumps that they would do the job and, accordingly, orders were placed for the Sewaren pumps. Operating experience to date has confirmed the expectation of satisfactory operation.

The traditional steam-driven auxiliary oil pump of the turbine

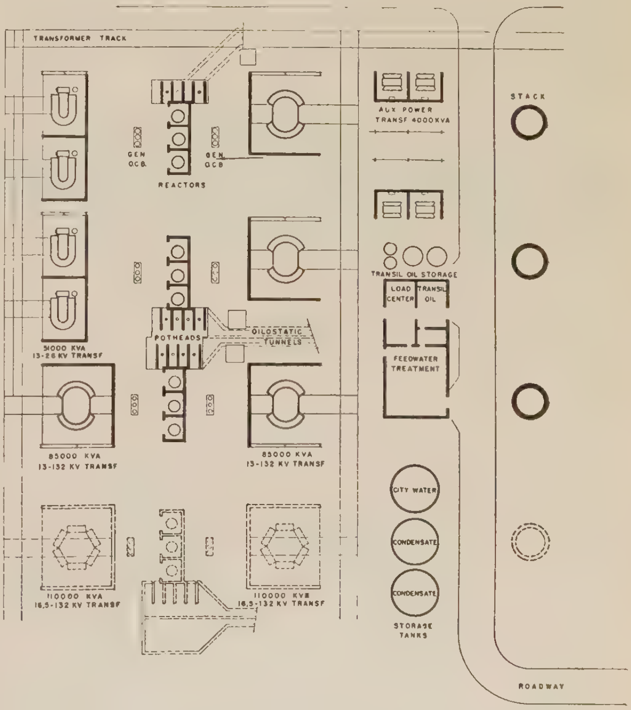


FIG. 24 PLAN OF TRANSFORMER YARD—SEWAREN

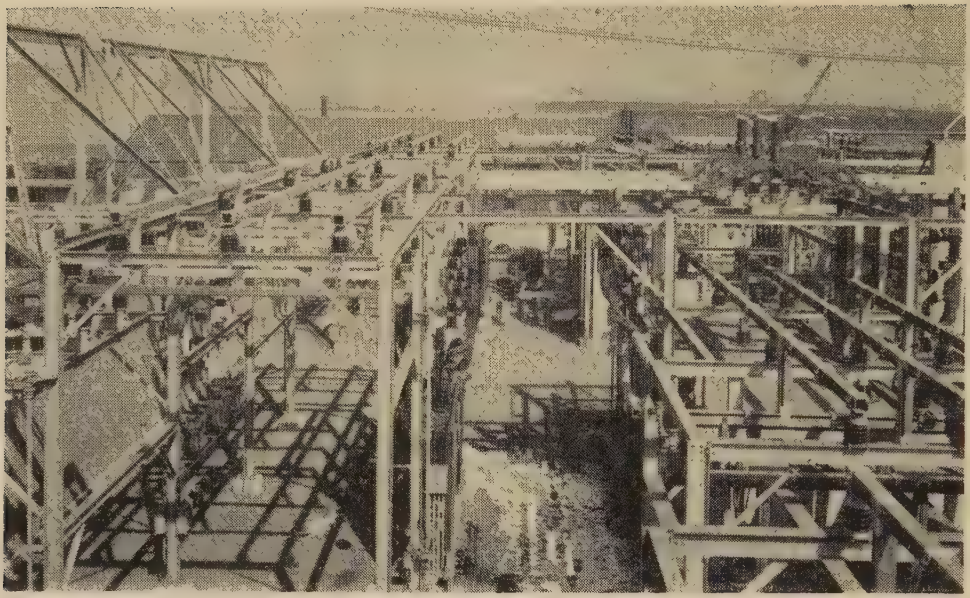


Fig. 25 13-Kv Aluminum Buses at Sewaren

generator was replaced by a full-capacity alternating-current motor-driven pump, backed up by a smaller direct-current motor-driven pump operating from the station battery. This latter pump has sufficient capacity to bring the turbine-generator to a stop if both main and auxiliary pumps should fail.

The arrangement of auxiliaries at the lower operating level of Sewaren is shown in Fig. 21. This plan, which includes only the first two units, is typical for the second pair.

Fuel Handling and Storage

Sewaren is designed for normal operation with water-borne fuel. The plan of the property, Fig. 22, shows the coal-unloading tower and breaker house, where the coal is sent either to the bunker or to the swing-boom tower for storage by tractor and carry-all. Both the dock and the unloading tower are designed so that in the future the unloading tower could be mounted on wheels and moved along the dock. Coal arriving by rail or reclaimed from yard storage is dropped into track hoppers feeding a conveyer to the breaker house. A 125,000-bbl fuel-oil tank is located north of the main building.

Principal Electrical Features—Sewaren

The ultimate connections of the four Sewaren generators to the system are shown in Fig. 23. Each generator feeds into the midpoint of a reactor which takes the outgoing current into two channels. All switching is done between the generator and transformer at 13 kv, except for one 132-kv oil circuit breaker which ties together two lines, closing a transmission loop when the station is shut down. The 26-kv circuits feed the area immediately adjacent to Sewaren, while the 132-kv lines connect to the transmission system for utilization in more remote areas. Fig. 24 shows the arrangement of equipment in the transformer yard.

The 13-kv generator leads extend about 500 ft to the transformer yard in 8-in. oilostatic pipes in tunnels on foundation slabs carrying station power and control cables in one case, and transformer track supports in another case.

Placing the oilostatic cables in tunnels enabled the use of forced cooling, by means of which the copper cross section of the conductors was reduced from 12 to 6 million circular mils per phase, thus effecting a saving in cost and permitting the use of small easily handled cables. Forced air is used to cool the pipes in No. 1 and 3 tunnels. For generator No. 2, the oil is circulated in the oilostatic pipe by a pump and gives up its heat in an oil-water heat exchanger. The forced cooling is required only in hot summer weather. Each generator has twenty-four 750,000-circular-mil oilostatic cables, six in each 8-in. pipe.

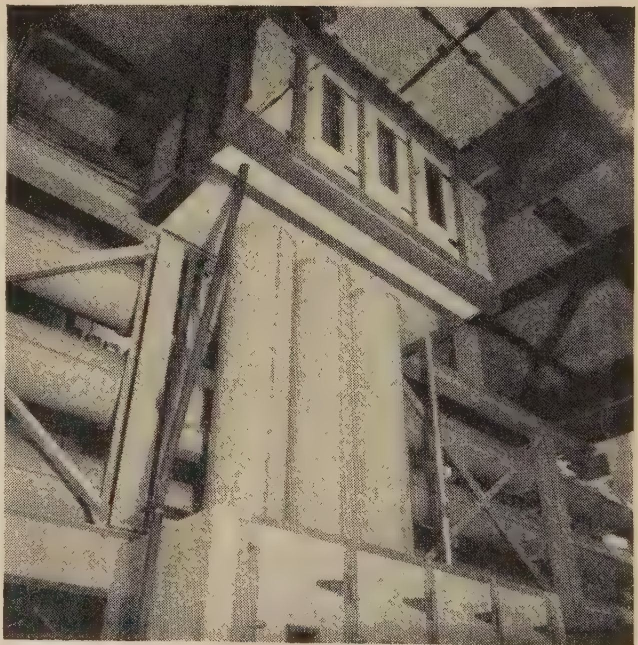


Fig. 26 Isolated Phase Bus Leads From Auxiliary Generator —Sewaren

The aluminum bus[3] was adopted for the transformer yard after a study showed considerable savings in its installation. Fig. 25 shows portions of the square 13-kv aluminum buses. At the left they are tubular and at the lower right they are formed from two angle sections. A large portion of the savings lies in the use of the

---

[3] "Welded Aluminum Bus in Outdoor Transformer Yard of Sewaren Generating Station," by D. M. Quick, Edison Electric Institute Bulletin, December, 1948.

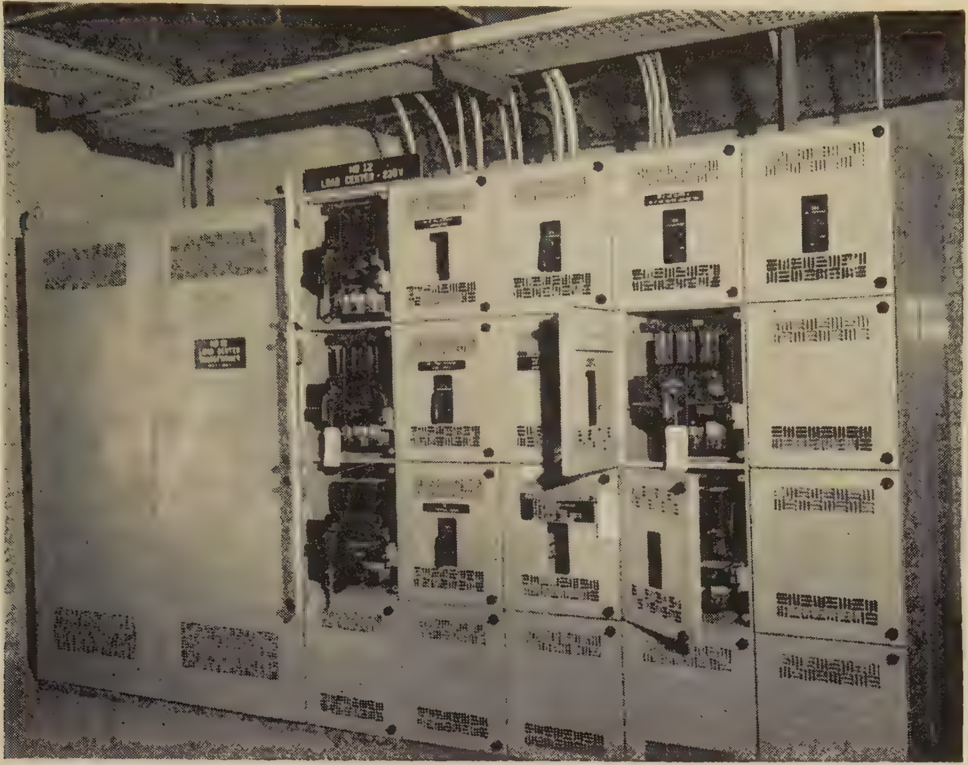


Fig. 27 220-Volts Load Center- Sewaren

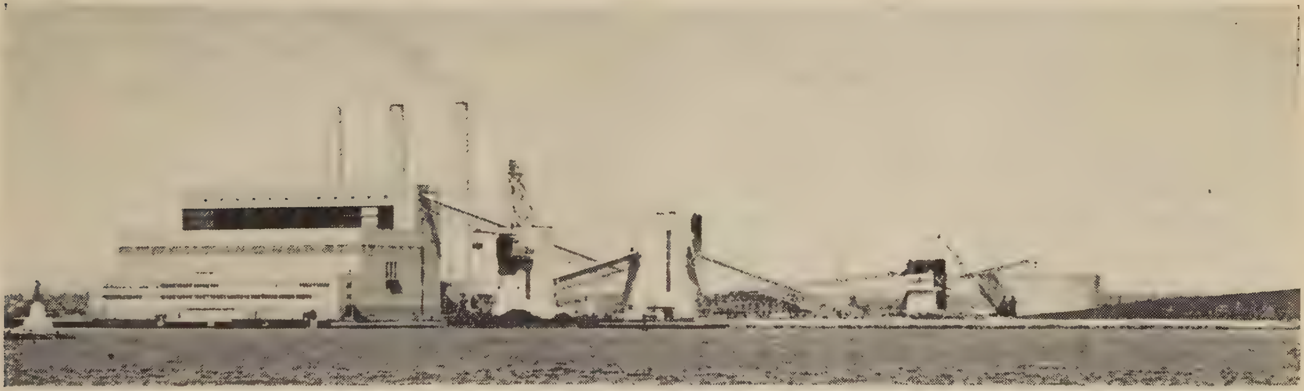


Fig. 28 Sewaren Generating Station

newly developed inert gas-shielded arc system of welding in lieu of the usual bolted connections. This is the first installation ever made using inert gas-shielded-arc welding on outdoor buses of such heavy cross section.

The isolated phase bus leads coming from the auxiliary generator to the 2400-volt service buses are shown in Fig. 26. A stovepipe bus comes from the main disconnecting switch to a potential transformer compartment and to the cross bus. Toward the right it terminates in a rectangular metal-clad enclosure feeding one of the 2400-volt auxiliary buses.

Motors rated at 200 to 2000 hp are supplied at 2400 volts; those between 25 and 200 hp at 440 volts; while those below 25 hp are fed at 220 volts. These latter motors and the station lighting are supplied by air-cooled transformer load center units located throughout the station, Fig. 27.

Totally enclosed motors have been used to the greatest extent practicable. In addition to the fan motors already mentioned, all motors for the coal-handling system, boiler feed pumps, and the circulating-pump motors of No. 3 and 4 units, are totally enclosed.

Buildings

The architectural treatment of the buildings is simple and functional, with strong horizontal lines resulting from continuous rows of sash and glass block, Fig. 28. At the permanent end of the turbine room there is a visitors' observation balcony which affords an outside view of the easterly end of the property and an inside view of the turbine room. The main entrance and lobby are at the permanent end of the service building. All the window sash are of aluminum, and the principal entrance doors are of stainless steel.

The turbine-room roof is supported by arched girders rather than by the conventional open-truss construction. The temporary end is covered with corrugated asbestos sheets secured to steel girts. The coal-handling buildings are enclosed with flat transite sheets.

LIGHTING

A high level of illumination is provided throughout the station in the belief that it will yield dividends in better operation, safety, and general housekeeping. In this connection, special attention was given to the turbine room, Fig. 19. Combination incandescent-fluorescent units are mounted flush against the bottom flange of the roof beams to form a continuous fixture containing sixteen 500-watt incandescent lamps with control lenses, and sixty-four 40-watt fluorescent tubes, about 10 kw of lighting load per beam. Relamping and cleaning are accomplished readily from the top of the crane.

SUMMARY

The objectives in the Sewaren design were straightforward simple arrangement which with central control would simplify operation and make for minimum man power per kilowatt; a reasonably efficient cycle that would not be obsolescent at too early a date; and a simple building construction involving a minimum of maintenance throughout its life.

ACKNOWLEDGMENT

The author gratefully acknowledges the assistance of Messrs. E. H. Snyder, H. Weisberg, and P. A. Salmon, all of the Public Service Electric and Gas Company.

Appendix

PRINCIPAL EQUIPMENT FOR NO. 1, 2, AND 3 UNITS AT SEWAREN

Turbine-Generators and Auxiliaries:

1 Turbine-generator (Westinghouse), having nameplate data as follows:

Steam turbine, rated 100,000 kw, maximum 109,500 kw; initial pressure 1500 psi, initial temperature 1050 F, exhaust pressure 1.5 in. Hg, rpm 3600, 32 stages.

Alternating-current generator, rated 121,176 kva, power factor 0.85, hydrogen pressure 30 psi, frequency 60 cycles, phases 3, voltage 13,100 to 14,500, star connected, speed 3600 rpm.

Auxiliary alternating-current generator, rated 9375 kva, power factor 0.80, air cooled, speed 3600 rpm.

2 Turbine-generators (General Electric) each having nameplate data as follows:

Steam turbine, rated 100,000 kw, 23 stages, speed 3600 rpm, maximum capacity 108,500 kw, steam pressure 1500 psig, temperature 1050 F, exhaust pressure 1.5 in. Hg abs.

Alternating-current generator, hydrogen cooled, 2 poles, 60 cycles, 3 phase wye-connected for 13,800 volts; rating at 15 psig hydrogen pressure 111,765 kva at 0.85 power factor, armature 4670 amp, field 1040 amp, exciter 250 volts. Capacity at 25 psig hydrogen 117,647 kva at 0.85 power factor; armature 4920 amp, field 1070 amp.

Auxiliary alternating-current generator, rated 9375 kva, power factor 0.80, air cooled, speed 3600 rpm.

3 Single-pass, divided water box, spring supported, welded steel-shell Foster-Wheeler condensers with storage-type deaerating hot well; one 50,000 and two 60,000 sq ft surface, and $^7/_8$-in-OD $\times$ No. 18 BWG aluminum brass and aluminum bronze tubes.

9 Low-pressure, 2-pass, closed feedwater heaters (Westinghouse), located horizontally in condenser neck.

3 Low-pressure, 4-pass, closed vertical feedwater heaters; and 12 high pressure, 2-pass, closed vertical feedwater heaters (Foster Wheeler Corp.)

3 Condensate coolers, using salt water for cooling (Graham Manufacturing Co.).

Steam-Generating Units and Auxiliaries:

3 Boilers (Combustion Engineering-Superheater, Inc.), each rated 850,000 lb per hr continuous, 1500 psi, 1050 F throttle steam conditions, 52,450 cu ft furnace volume, 15,080 sq ft boiler heating surface; vertically adjustable, tangential firing burners, continuous slag tap; 2-stage pendant, convection superheaters, 41,800 sq ft heating surface; continuous tube, counterflow, 39,600 sq ft economizer.

6 Ljungstrom regenerative air heaters (Air Preheater Corp.), 104,350 sq ft.

Superheater and economizer refractory walls, sectionally supported (M. W. Detrick Co., George Allen & Son).

3 Ash hoppers, 3 ash pumps and 1 dewatering bin, Allen-Sherman-Hoff Co.

Ductwork, Connery Construction Co.

Boiler and boiler gas-duct insulation, H. W. Porter Co.

Hot air duct insulation, Chas. S. Wood & Co.

Air soot blowers, automatic sequential operation, Diamond Power Specialty Corp.

9 Hardinge pulverizers, 19 tons per hr; 9 raw-coal feeders; and 9 mill exhausters, Foster Wheeler Corp.

6 Variable-speed forced-draft fans, driven through hydraulic couplings, American Blower Corp.

6 Air-cooled radiator for cooling oil of forced-draft-fan hydraulic couplings, Young Radiator Co.

6 Induced-draft fans, 2 driven through hydraulic couplings, 4 louvre controlled, American Blower Corp.

2 Fan-cooled radiators for cooling oil in the induced-draft-fan couplings, General Electric Co.

3 Stacks, double shell, radial brick, 225 ft high, 15 ft diam at top, M. W. Kellogg Co.

Zeolite water softening system, Worthington Pump & Machinery Corp.

3 Evaporators, 30,000 lb per hr each (2 Foster Wheeler Corp.; 1 Griscom-Russell Co.).

3 Evaporator deaerators, Worthington Pump & Machinery Corp.

Coal-Handling Equipment:

Unloading tower and barge puller, McKiernan-Terry Corp.

Bradford breaker, Pennsylvania Crusher Co.

Conveyors, traveling tripper, and swing boom, Robins Conveyors, Division of Hewitt-Robins Inc.

Conveyor belts, Thermoid Rubber Co.

Magnetic separator, Ding's Magnetic Separator Co.

Weightometer, Merrick Scale Manufacturing Co.

12 Automatic coal scales, Richardson Scale Co.

Raw-coal hoppers and stainless-steel chutes, Heilman Boiler Works; Allied Steel Products Corp.

Gates in raw-coal chutes and at bunker outlets, Stock Engineering Co.

2 Tractors, bulldozers and carryalls, Smith Tractor Co.; Eastcoast Equipment Co.

Dust-Handling Equipment:

Dust collectors, multiclone-Cottrell, Western Precipitation Corp.

Fuller-Kinyon dust-conveying system, Fuller Co.

Redler conveyers between precipitators and dust-pump hopper, Stephens-Adamson Manufacturing Co.

Variable-speed drives, Reeves Pulley Co.

Screw conveyers between economizers and dust-pump hopper, Link Belt Co.

Star feeder valves, precipitator, and economizer hoppers, United Conveyer Corp.

Dust-storage silo, Heilman Boiler Works

Pumps:

6 Vertical, mixed-flow, 47,000-gpm, 345-rpm circulating-water pumps, Foster Wheeler Corp.

6 Rotating-plunger-type, single-stage, vacuum pumps, 550 cfm at 1.5 in. Hg, Kinney Manufacturing Co.

6 Vertical, centrifugal 4-stage, re-entry type condensate and booster pumps; 2090 gpm at 517 ft head, 884 rpm, Worthington Pump & Machinery Corp.

9 Horizontal, centrifugal 8-stage, 1115-gpm at 4,680 ft head, 3465-rpm boiler feed pumps, Worthington Pump & Machinery Corp.

9 Variable-speed hydraulic couplings driving boiler feed pumps, rated 2000 hp at 3550 rpm, American Blower Corp.

4 Vertical, DeLaval, 2500-gpm at 250 ft head, 1760-rpm salt-water service pumps, Turbine Equipment Co.

3 Ash sluice pumps, 2000 gpm at 25 ft head, Worthington Pump & Machinery Corp.

1 Centrifugal fire pump, 1600 gpm, Gould Pumps, Inc.

1 City-water booster pump, 500 gpm, Gould Pumps, Inc.

3 Vertical, DeLaval, rotary-screw, fuel-oil pumps, 140 gpm at 325 psi, Turbine Equipment Co.

2 Pumps for acid-cleaning system, 1000 gpm at 70 ft head, Worthington Pump & Machinery Corp.

9 Sump pumps, Quimby Pump Division, H. K. Porter Co. Inc.

2 Condensate storage tank pumps, Ingersoll-Rand Co.

3 Evaporator feed pumps, Gould Pumps, Inc.

1 High-pressure chemical feed pump, Worthington Pump & Machinery Corp.
4 Continuous chemical feed pumps, Milton Roy, Inc.
2 Condensate return pumps, Worthington Pump & Machinery Corp., and Federal Pump Corp.
1 Circulating pump for oilostatic cable cooling, Sier-Bath Gear & Pump Co., Inc.
1 Laboratory vacuum pump, Schutte & Koerting Co.
2 Fuel-oil-reclaiming pumps, Sier-Bath Gear & Pump Co., Inc.

Air Compressors:
4 Horizontal, single-stage, 150-cfm at 100 psi carbon-ring-control air compressors, Ingersoll-Rand Co.
1 Horizontal, duplex, 2-stage, 660-cfm at 125 psi station-service air compressor, Ingersoll-Rand Co.
2 Rotary, 2-stage, 1035-cfm at 120 psi soot-blower air compressors, Fuller Co.
2 Horizontal, single-stage, 1000-cfm, 120 to 500 psi high-pressure soot-blower air compressors, Ingersoll-Rand Co.
6 Dual-tower silica-gel air dryers for control air, Dehydraire Corp.

1 Air receiver, 1000 cu ft, 500 psi, A. O. Smith Corp.
3 Air receivers, 60 cu ft, 250–320 psi, American Locomotive Co.

Instruments and Controls:
Three-element control of boiler feed pump speed, Bailey Meter Co.
Automatic combustion-control equipment, Hagan Corp.
160-point bearing and superheater-tube temperature recorders; and 90-point miscellaneous temperature recorder, Leeds & Northrup Co.
Automatic control of superheater by-pass damper, Leeds & Northrup Co.
Condenser-hot well-level control equipment, Hagan Corp.
Automatic fuel-oil pressure control, Hagan Corp.
Yarway remote boiler-drum-level indicators, Yarnall-Waring Co.
Boiler meters, Bailey Meter Co.
Clear and bi-color gage glasses for boiler drums, Diamond Power Specialty Corp.
Miscellaneous gage glasses, Ernst Water Column & Gage Co.
Boiler-draft and pressure gages, Republic Flow Meters Co.
Ashcroft Duragages, Hajoca Corp.
CO_2, pH, temperature, and conductivity recorders, Leeds & Northrup Co.
Industrial thermometers, Taylor Instrument Co.

Piping:
Main steam and high-pressure boiler feed piping, M. W. Kellogg Co.
Intermediate and low-pressure piping, Midwest Piping and Supply Co.; Cornell & Underhill
Circulating-water 48-in. reinforced concrete pipe, Lock Joint Pipe Co.
Strainers, Andale Co.; Leslie Co.
Insulation, Chas S. Wood & Co.

Valves:
Steel gate and check valves, high pressure, The Lunkenheimer Co.
Steel gate and check valves, Wm. Powell Co.; Walworth Co.
Safety valves, Foster Engineering Co.; Manning, Maxwell & Moore Inc.
Bleeder check valves, Schutte & Koerting Co.
Boiler blowdown valves, Yarnall-Waring Co.
Regulating valves, Bailey Meter Co.; Fisher Governor Co.; Swartwout Co.; Schutte & Koerting Co.
Pressure relief valves, Manning, Maxwell & Moore Inc.; Farris Engineering Co.; Foster Engineering Co.; Crosby Steam Gage & Valve Co.
Iron valves, Crane Co.; Chapman Valve Manufacturing Co.; Darling Valve and Manufacturing Co.; Walworth Co.
Butterfly valves for circulating water, Henry Pratt Co.

Electrical Equipment:
9 Reactors, single-phase, mid-tap, 5000 amp, General Electric Co.
4 Transformers, 85,000 kva, 13.2–132 kv, forced oil-forced air, Westinghouse Electric Corp.
4 Transformers, 51,000 kva, 13.2–26.4 kv, forced air, General Electric Co.
4 Station power transformers, 4000 kva, 26.4–2.4 kv, Moloney Electric Co.
2 Station power transformers, 1500 kva, 2400–440 volts, Allis-Chalmers Mfg. Co.
Lightning arresters, 15 kv (6 General Electric Co.; 3 Westinghouse Electric Corp.)
Oil circuit breakers, 15, 34.5, and 132 kv, Westinghouse Electric Corp.
Main generator leads of oilostatic cables, Okonite-Callender Cable Co. Inc.
Auxiliary generator-lead housings, Railway and Industrial Engineering Co.
Disconnecting switches 7.5, 23, 34.5, 132 kv, Railway and Industrial Engineering Co.; Delta Star Electric Co.; Pringle Manufacturing Co.
228 Transformer cooling fans, Diehl Manufacturing Co.
Metal-clad switchgear, 2400 and 440 volts, Westinghouse Electric Corp.
Load centers, 220 volts, Westinghouse Electric Corp.; I-T-E Circuit Breaker Co.
Storage batteries, 125 and 250 volts, Electric Storage Battery Co.
Electric motors, 220, 440, and 2400 volts, 3-phase, 60 cycles, Westinghouse Electric Corp.; Allis-Chalmers Manufacturing Co.; General Electric Co.
Mulsifyre fire-protection equipment, Grinnell Co.

Miscellaneous:
6 Sluice gates, operated by air motors, Coldwell-Wilcox Division, Krajewski Pesant Manufacturing Corp.
5 Traveling water screens, basket type, Link-Belt Co.
1 Traveling water screen with continuous screen cloth, Stephens-Adamson Mfg. Co.
Chlorination equipment, Wallace & Tiernan Products, Inc.
150-ton turbine-room crane, Whiting Corp.
2 Boiler-house elevators, Otis Elevator Co., Westinghouse Electric Corp.
1 Service-building elevator, Otis Elevator Co.
Ventilation and air conditioning, Buensod-Stacey Inc.
Heating boiler, Cleaver-Brooks Co.
Test Department furniture, Laboratory Equipment Co.
DeLaval turbine and transil oil-purifying equipment, Turbine Equipment Co.
Structural steel: Main building, Lehigh Structural Steel Co.; Turbine-generator supports, American Bridge Co.; Coal-handling structures, Morris, Wheeler & Co.
125,000-bbl fuel-oil tank, Bethlehem Steel Co.
Fuel-oil tank heaters, Griscom-Russell Co.
Fuel-oil pressure heaters, American Locomotive Co.
3 Water-storage tanks, 125,000 gal each. Bethlehem Steel Co.
Equipment insulation, Chas. S. Wood & Co.

NOTE: List of principal equipment for No. 1 unit, Essex Generating Station, was published in October, 1948, issue of *Power*.

Designers, Electric Engineering Department, Public Service Electric and Gas Company.
Builders, United Engineers & Constructors Inc.
Architectural consultants, Walker & Poor.

Discussion

C. B. Campbell.[4] The author has presented a most interesting report on the background experience of his company, culminating now in the modern Sewaren generating station. Certainly this is a plant whose continuing performance will be watched with great interest by all who are concerned with steam-power-generating problems.

The author refers to the installation of new equipment to modernize Essex Station as a definite step in the development of many Sewaren innovations and principles. Actually, it seems, at the expense of greater length and detail, that he could have included the progressive developments previously made by his company in its Burlington, Essex, and Marion generating plants as also having contributed to the latest accomplishment. Taken in their entirety, this whole program presents an excellent chronological picture of the progress in American steam-power-plant practice over the past 15 years.

Table 2 of the paper lists materials used in pressure parts exposed to the highest-temperature steam for the specific units indicated by column headings. It is believed there may be some misunderstanding of these data on the part of those not intimately acquainted with details of design of the equipment. We would like, therefore, to amplify this matter somewhat as regards Sewaren turbine, unit No. 1.

---

[4] Manager of Engineering, Steam Division, Westinghouse Electric Corporation, Philadelphia, Pa. Mem. ASME.

While the main steam-piping system leading to the turbine is of 3 per cent Cr 1 per cent Mo alloy as listed, the inlet steam piping connecting the separately supported turbine control-valve chests to the several first-stage nozzle groups is of 17–20 per cent Cr, 9–13 per cent, Ni, Cb-stabilized steel. Higher alloy was selected for these inlet pipes in order to reduce wall thickness and the length of loop necessary to care for expansion and relative displacement of the terminal connections.

In so far as the turbine is concerned, highest-temperature steam, that is, at 1050 F, is confined only to the throttle valves, control-valve chests, inlet piping, nozzle chambers, and nozzle blocks. Steam does not come into contact with the turbine cylinder proper until heat has been extracted by work performed in the first turbine stage, and its temperature even at maximum loading has decreased to approximately 970 F. It is this reduction from the maximum temperature of the steam supplied which permits the use of $2^1/_4$ per cent Cr, 1 per cent Mo alloy for the inner turbine cylinder.

The throttle and control-valve bodies and bonnets are of steel forgings of the composition shown in Table 2 under "Turbine Valves." Of the parts enumerated in the preceding paragraph as being subjected to 1050 F steam, castings were used only for the relatively small individual nozzle chambers. These castings are of 16–18 per cent Cr, 11–13 per cent Ni, 1.75–2.75 per cent Mo, with Cb stabilizer.

W. L. Chadwick.[5] The author has presented a valuable paper on an outstanding power station, a station which it was the writer's good fortune to inspect early in 1949. The Public Service Electric and Gas Company and its engineers are to be complimented on their courage and foresight in (1) pioneering the use of large units operating on steam at 1050 F, (2) the high efficiencies obtained, (3) building semioutdoor boilers in a comparatively severe climate; and further, for their ingenuity in simplifying plant layouts, centralizing controls, and in eliminating so many of the customary steam auxiliaries. Doubtless many of the new developments which are being field-tested and proved in Essex and Sewaren will soon be accepted standard practices.

The increases in efficiency which the author reports are impressive, the No. 4 unit at Sewaren being expected to gain nearly 10 per cent in efficiency when compared with the Essex unit, which latter unit is an efficient generator by any present standard. This efficiency represents about 665 kwhr per bbl of oil on the Pacific Coast base of 6,250,000 Btu per bbl. Some of the newer stations being built throughout the country are designed for as much as 100 kwhr per bbl less than this 665. The use of eight feedwater heaters in the cycle appears somewhat elaborate, but the efficiency and the station cost seem to justify that use. The use of triple-flow turbine exhausts to offset the higher exhaust losses usually attending 3600-rpm designs is commendable.

The clean simplicity of Sewaren's unit design described by the author not only reduced first cost, but should make for considerably greater operating reliability through elimination of headers and manifolds with their multitude of fittings and valves, each of which is a potential source of an outage. The use of semioutdoor boilers on the East Coast at a latitude of 41° is an interesting and bold step, but one which seems well-justified by a saving of $1,000,000 in construction costs of the first three units. The structure arrangement, which encloses the front and one side of each boiler, is ingeniously planned to give protection to weather-vulnerable equipment at minimum building costs. It is also notable that this efficient station is being built for a total cost of $132 per name-plate kilowatt.

The advances in simplifying auxiliaries are also significant. The development of air-cooled bearings and fluid couplings, and of totally enclosed motors to facilitate outdoor operation is interesting and creditable. The use of motor-driven vacuum pumps to replace the usual condenser jets puts to test the results of considerable recent discussion and development with several valuable advantages. The air soot blowers and the consequent elimination of many auxiliary steam lines also tests in a major station other recent developments and is a progressive step.

The Sewaren control system is efficient and convenient. The contrast which the author notes between the 521 sq ft of panel at one location at Sewaren and the 695 sq ft of control panel at five separate locations for a comparable unit at the Burlington Station is an interesting indication of the advantages of central control. The Sewaren design, which groups the control of each pair of units into a single control room is practical, convenient, and effective in reducing operating labor.

Electrically, Sewaren is ingeniously fitted to special system requirements.

The use of main turbine-driven house generators for auxiliary power appears to be contrary to the current trend. Most of the newer steam stations supply auxiliaries for each unit as a separate group, as at Sewaren, but use transformer banks, usually directly connected to the generator leads, as auxiliary power sources.

Louis Elliott.[6] To one who knows the author of this paper and holds him in high regard, the description of Sewaren and its basis of design exemplify his tireless application and his high competence. The writer is sure he will take pride, through the years to come, in the great plant to which he has given so much care and work, from the initiation of the project to the completion of construction.

While the design incorporates many modern refinements and new developments, the unit cost reported—$132 per kw—is moderate.

The writer was interested in the reported reduction in total construction cost, by adoption of semioutdoor steam generators. The saving is given as $1,000,000 for three units; this is at the rate of roughly $3 per kw—which happens to be approximately the same unit saving as has been observed in other plants, achieved by similar reduction in structures. It is gratifying to see the adoption of this type of plant construction by a number of utilities operating properties in the East and Middle West, where severe winter conditions are experienced, as well as by companies in the West. The economy achieved is worth while, and the design should not entail serious difficulties in operation and maintenance.

The next step in reduction of structures in steam-plant construction involves use of an outdoor gantry crane, tramming over a low structure housing the turbine generator. This is figured to save around $3 per kw, additional, in plant cost. The third step calls for the installation outdoors of the turbine and considerable other equipment, with housing limited to that for control room and for protection of certain delicate equipment. The saving as compared with a fully housed plant may be of the magnitude of $10 or more per kw.

A recent example of the third step in economy is a gas-burning plant just now going into operation in the South, with two 66-mw turbine generators each served by one steam generator. The executives and engineers on this property are thoroughly in accord with the adoption of the design, and with its substantial savings.

---

[5] Manager, Engineering Department, Southern California Edison Company, Los Angeles, Calif. Mem. ASME.

[6] Consulting Mechanical Engineer, Ebasco Services Incorporated, New York, N. Y. Mem. ASME.

Many of the outdoor plants have been in the South, but a number of stations involving the second step in reduction of structures have been installed and operated in locations experiencing winter conditions.

Plants designed during the past 10 years, incorporating various types of outdoor and semioutdoor construction, have aggregated between 3,000,000 and 4,000,000 kw in capacity. The 150 to 200 plant-years of successful operation of these stations have demonstrated the practicability as well as the economy of that character of design.

C. C. Whelchel.[7] The author clearly explains the evolution of the Sewaren design over a brief but significant period starting with the Essex unit and ending with the reheat Sewaren unit No. 4. Through all, economic considerations have been a keynote as well as consideration of reliability, always aiming at the designers' goal—the lowest annual cost. This entails not only the most careful consideration of initial capital and operating cost, but also efficiency.

The semioutdoor boiler arrangement has proved to be a natural point of savings here in California and has been used for this reason for many years, but it would seem to present problems, although not insurmountable, where winters are more severe. Serious operating difficulties were encountered this last winter at one of our plants when unprecedented cold weather resulted in the freezing of exposed control and gage lines. This brief contact with the freezing problem has instilled in us considerable respect for the problems which must have been encountered in evolving the semioutdoor design for a colder section of the country. For example, a report has reached us of an experience of a boiler-water level gage freezing on an operating boiler due to a severe building air leakage. We would be interested in a brief résumé of methods used to avoid cold-weather difficulties at Sewaren.

As pointed out by the author, the centralization of control results in better co-ordination of operations and reduction in operating attendance. It is apparent that this may have been carried to a practical limit. The designer must be aware that as long as equipment is functioning properly few operators are required. The real test of adequacy is during and immediately after disturbances.

The effort expended to minimize required make-up is believed to be well-placed. It is probable that while this will show returns, as far as annual operating costs are concerned, little if any benefit can be derived in the sizing of make-up equipment.

While the omission of the deaerating heater from the Sewaren cycle may have simplified plant design and operation, and in addition, resulted in a lower initial cost, it would appear that the hazard of feed-pump seizure due to flash may be introduced upon sudden load changes or condensate-pump failures. This problem has been solved satisfactorily in the case of pumps fed from the deaerating heater.

Author's Closure

The author thanks Mr. Campbell for the information clarifying Table 2.

With regard to Mr. Chadwick's comment about main turbine-driven house generators for auxiliary power, the primary objective, as stated in the paper, was to provide the auxiliaries with a direct source of power free from outside disturbances. This is particularly important with pulverized-coal firing where there is danger of the fire being extinguished due to slowing down of the pulverizing equipment with a system voltage drop and relighting when the voltage is restored. We have had a number of trip-outs of high-tension transmission circuits due to dancing conductors caused by ice formation. In some instances these trip-outs have resulted in the separation of parts of the system and the loss of the main generators and their transformer-fed auxiliaries at our older stations. In using a shaft-end auxiliary generator supplying auxiliary buses not tied in with the main system nor the auxiliary system of any other unit, we expect our auxiliaries to operate undisturbed through the most severe disturbances. We realize, of course, that the ultimate in dependability is accomplished by a separate house turbine. The complication involved, however, is too great for the advantages gained. Also, our policy is to drop load rather than allow frequency to diminish.

An important consideration was the size of the turbine-generator unit. At the time that this station was designed, the upper limit of the rating of a single-shaft turbine-generator unit was determined by generator size. Thus the rating of the unit was increased by generating auxiliary power in a separate, shaft-end machine.

In accordance with Mr. Whelchel's suggestion, the following is a brief description of cold-weather precautions for outdoor equipment. The boiler drums, superheater outlet header, gage glasses, and safety valves are placed forward of the boiler and inside the boiler-operating aisle. All stairways serving the outdoor boiler platforms are indoors. Silica-gel driers are used to remove moisture from the compressed air which operates control devices and other similar equipment. Soot-blower mechanisms and other control devices which are vulnerable to icing, are provided with covers or are placed in individual enclosures. Draft fan bearings are cooled by air flow induced by suction from the induced-draft side of the air heaters. Low pour-point lubricating oil of a relatively narrow viscosity range is used in bearings and hydraulic couplings, the same grade being used in winter and summer. Heat generated by slip of the hydraulic couplings of the forced-draft fans is transferred to the air entering the fans by means of radiators mounted at the fan inlets. In the case of the hydraulic couplings of the induced draft fans of No. 1 Unit, radiators with motor-driven fans are used for the same purpose. Air heater bearings are cooled with air supplied by a motor-driven fan.

With reference to Mr. Whelchel's comment about the omission of the deaerating heater, the feed pumps are protected against the hazard of seizure upon sudden load changes by a large storage hot well under the condenser which operates as a surge tank. This hotwell receives water from the head tank via the condenser deaerator when the level falls below a certain point. The system is protected against condensate pump failure by the same head tank which holds a five-minute supply of condensate available to the feedpump suction through a check valve which is held closed by normal pressure from the condensate pump.

---

[7] Chief, Division of Steam Engineering, Pacific Gas and Electric Company, San Francisco, Calif. Mem. ASME.

Operating Characteristics of the 100,000-Kw Essex Turbine Generator

By STANFORD NEAL[1] and VINAL S. RENTON[2]

On December 18, 1947, the largest 3600-rpm tandem-compound steam-turbine generator ever built up to that time was placed in service at the Essex Generating Station of the Public Service Electric and Gas Company, Newark, N. J. Since it was the first of a new class of turbine for both the manufacturer and the operating company, and since the throttle steam temperature was 1000 F, extensive tests were made to determine practical operation procedures for the new machine. These tests indicate that, even with the high steam conditions of 1250 psig this 1000 F turbine generator can safely withstand starting and loading rates higher than those normally used. The operating instructions for this machine can be revised to take advantage of these characteristics. Additional data are needed for wider application to other machines and designs.

Introduction

A TURBINE generator must be designed to operate safely and efficiently for many years under conditions of high pressures, high temperatures, and large centrifugal forces. Within 14 ft of the casing length of this particular turbine, the steam pressure changes from 1250 psig to 0.5 psia, and the corresponding temperatures change from 1000 to 80 F. The highest circumferential speed is 1393 fps.

With careful design, a turbine generator can be constructed to operate satisfactorily at a constant load under these severe conditions, but rapidly changing loads or temperatures impose stresses higher than occur at constant load. If all conditions of starting, load change, and steam-temperature change were imposed very slowly, the additional stresses due to changing conditions would be insignificant compared to those of constant load. Since very slow changes are not practical, it is necessary to determine the effects of rapid load and temperature changes, design the machine to withstand them, and then to specify, where possible, safe operating procedures during such changes.

Previous operating experience with turbine generators has shown that, without proper control, the length of starting time and rapid load and temperature changes may cause mechanical damage to the turbine or generator. To illustrate the importance of good operation, several of the extreme cases are mentioned here.

Both the turbine casing and shaft may bow because of nonuniform cooling after unloading, and the shaft may rub the labyrinth packings during the subsequent start and bow still more. The resulting unbalance can increase to such an extent that vibration may cause damage if the machine is handled improperly. This trouble has been largely prevented in later machines by rotating the shaft at very low speeds as the machine cools or is heated in preparation for starting.

Since the rate of heat transfer to the various parts is greatly different, crushing of internal fits and surfaces may occur, and the resulting stresses may permanently distort the parts.

When the rotor and stationary parts expand at different rates, mechanical contact may occur and wear the rubbing strips which minimize steam leakage. No serious mechanical damage will result from rubbing strip wear.

An infrequent but unusually severe condition exists during emergencies when water slugs are carried over from the boiler. Thrust-bearing failures and permanent distortions may result.

To reduce the possibility of such troubles, starting and loading rates have, in the past, been determined by experience. The tests described here, and recent tests of other investigators[3] indicate that, in some cases, those procedures of starting and loading are unnecessarily conservative, and may even be harmful. An adequate theory of starting and loading effects, substantiated by test data, should permit the establishment of more realistic operating procedures. Such information is important both to designer and operator.

Description of Turbine Generator

Since the mechanical description of the machine has been reported separately,[4] only a few details will be mentioned here. The semisection of the turbine is shown in Fig. 1, and the outline of the turbine and generator in Fig. 2. Figs. 3 and 4 show the inner and outer high-pressure shells. The latter three figures also show the location of several of the instruments used in this investigation.

The tandem-compound machine has a nominal rating of 100,000 kw, and the steam conditions are 1250 psig and 1000 F. The high-pressure section has 19 stages with extraction connections from the 7th, 10th, 13th, 16th, and 19th stages. The double-flow low-pressure section is on the same shaft and has 5 stages with extraction connections from each of the two 20th, 21st, and 22nd stages.

The main generator is rated 111,765 kva at 0.85 power factor with 15 psig hydrogen pressure and 117,647 kva at 0.85 power factor with 25 psig hydrogen pressure. The house or auxiliary generator is air-cooled and rated 7500 kw at 0.80 power factor. Separate motor-driven exciters supply the field electrical requirements.

The turbine and two generators are coupled solidly to one shaft. The unit has an over-all length of 77 ft $7^1/_2$ in., and weighs 1,120,000 lb.

Objects of Test

It was expected that there would result criteria for optimum operation with respect to safety, economy, and flexibility and, further, procedures for operation during emergency conditions.

---

[1] Section Engineer, Steam Turbine Engineering Division, General Electric Company, Schenectady, N. Y. Mem. ASME.

[2] Engineer, Electric Engineering Department, Public Service Electric and Gas Company, Newark, N. J. Mem. ASME.

Contributed by the Power Division and presented at the Semi-Annual Meeting, San Francisco, Calif., June 27–30, 1949, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 49—SA-28.

[3] "Quick Starting of High-Pressure Steam-Turbine Units," by J. C. Falkner, R. S. Williams, and R. H. Hare, Trans. ASME, vol. 70, 1948, pp. 201–209.

[4] "The Design of Sewaren Generating Station and No. 1 Unit at Essex Station, Public Service Electric and Gas Company," by F. P. Fairchild, published in this issue of the Transactions, pp. 247–266.

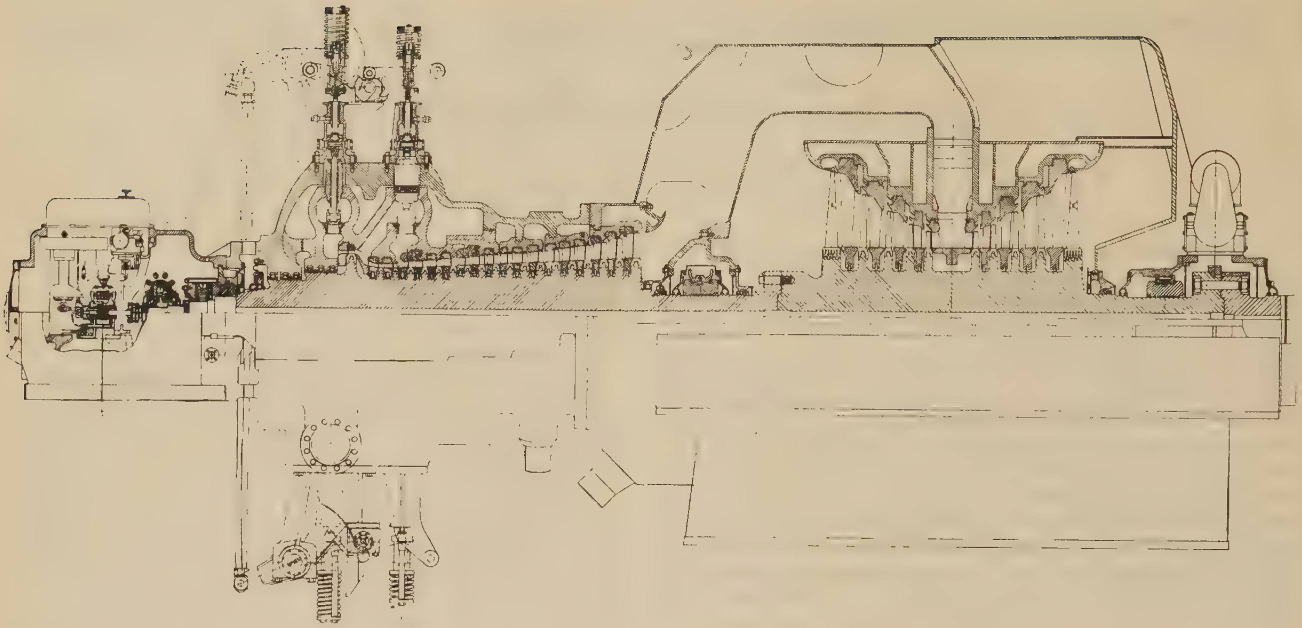


Fig. 1 Semisection of Unit No. 1 Turbine at Essex Generating Station of Public Service Electric and Gas Company

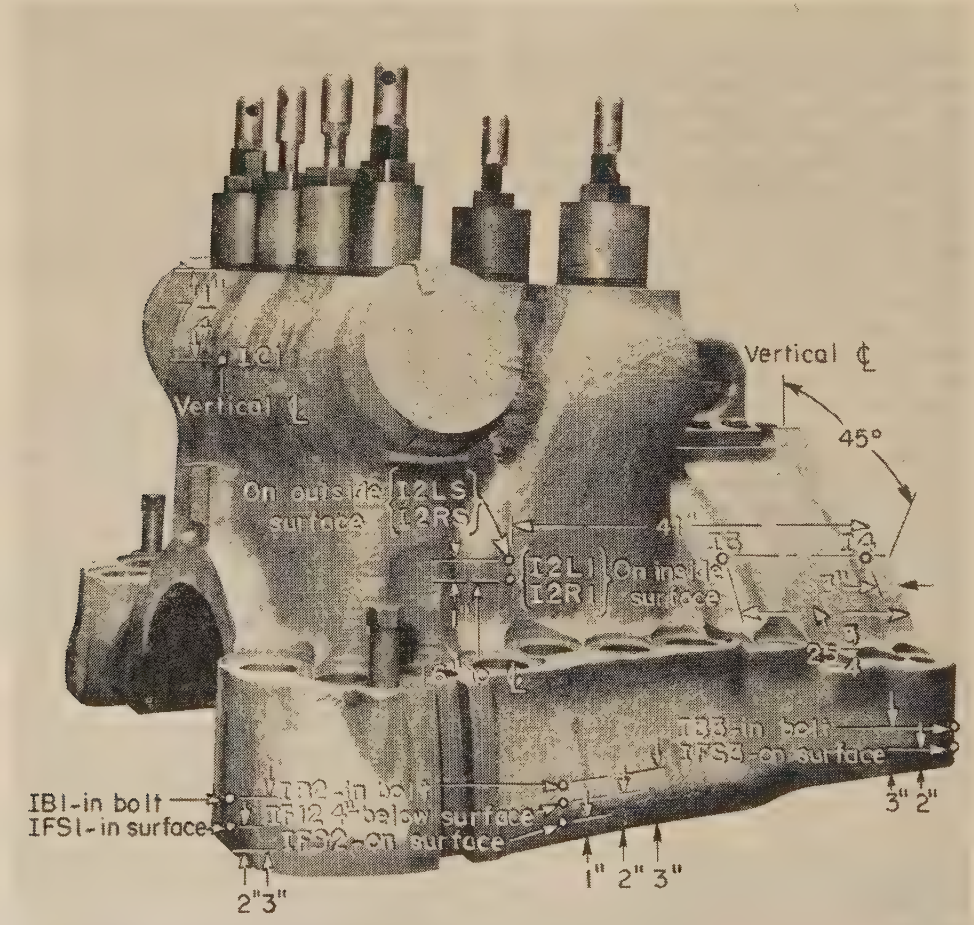


Fig. 3 Location of Thermocouples on Upper Half of Inner Shell of High-Pressure Turbine
(Nomenclature is described in the Appendix.)

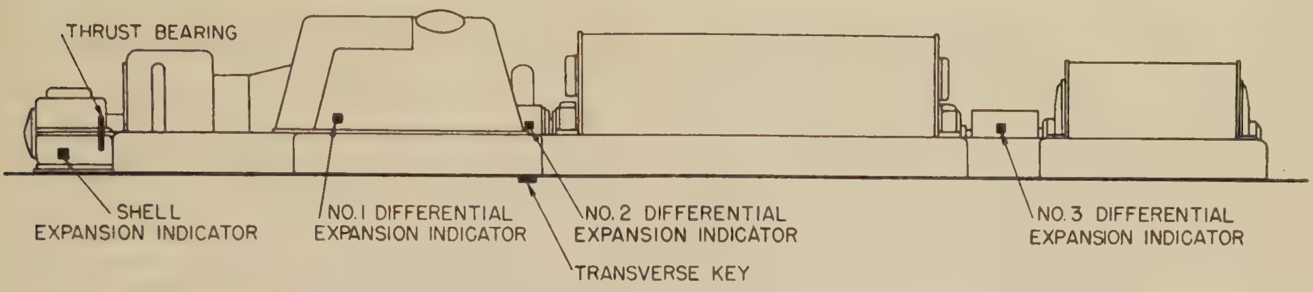

THRUST BEARING
SHELL EXPANSION INDICATOR
NO.1 DIFFERENTIAL EXPANSION INDICATOR
NO.2 DIFFERENTIAL EXPANSION INDICATOR
TRANSVERSE KEY
NO.3 DIFFERENTIAL EXPANSION INDICATOR

FIG. 2 OUTLINE OF TURBINE AND GENERATOR SHOWING LOCATION OF EXPANSION INDICATORS

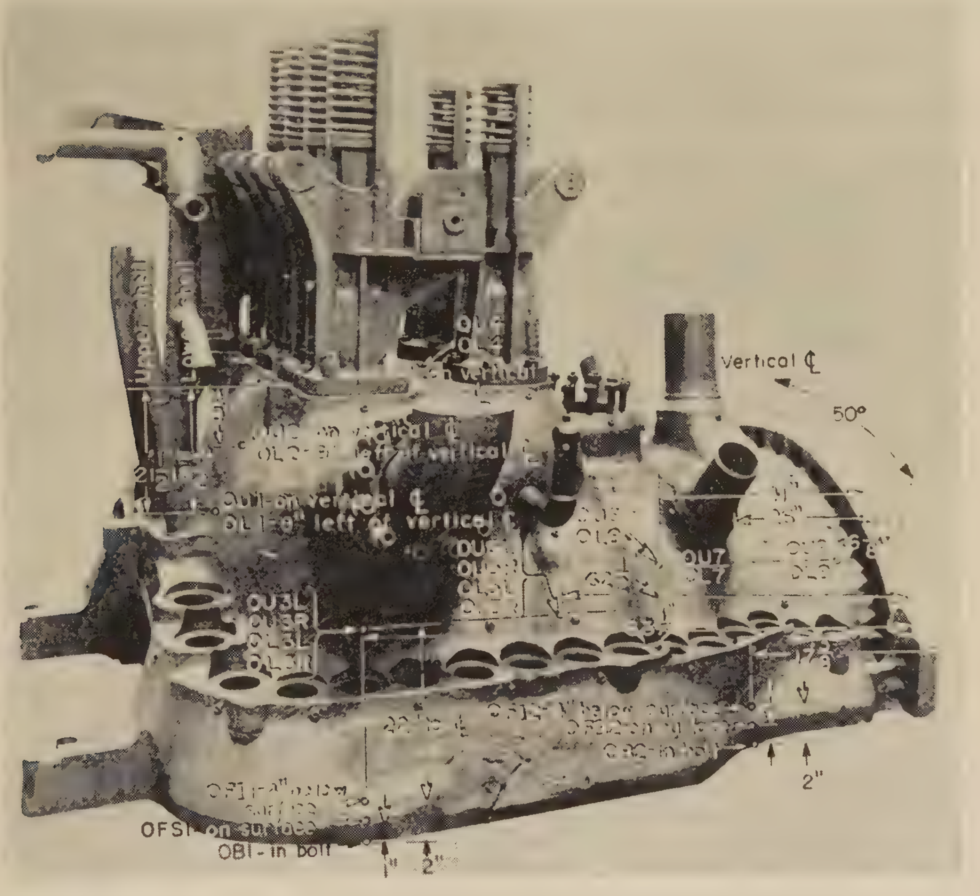


FIG. 4 LOCATION OF THERMOCOUPLES ON THE OUTER SHELL OF THE HIGH-PRESSURE TURBINE
(Only lower half is shown. Nomenclature is described in the Appendix.)

Particular attention was directed toward quick starting and high rates of load change, since it is sometimes desirable to start and load or to unload a turbine generator very rapidly during emergency conditions, and since it is economical in man power and fuel to reach high loads as quickly as possible from a start.

The specific objects of the test program were to determine the following:

1 Steady-state conditions of operation.

2 Effect of length of starting time.

3 Effect of changes of load and throttle steam temperature.

4 Effect of emergency operation, such as load dump and fast pickup of load.

5 Effect of preheating the generator field before or during starting. The generator-starting tests are to be reported separately.[5]

OPERATION DURING TESTS

The operation of the turbine and generator during the tests followed normal procedures with the only exceptions of starting time, loading rate, throttle steam temperature change, and generator heating as specified in the test program. The tests were made at the normal running clearances which have not been changed since the unit was put in operation.

The performance of the unit was observed closely, and at no time was there evidence of any distress or unusual operation.

---

[5] The generator preheating tests are expected to be the subject of an AIEE paper by F. W. Gay and H. D. Taylor.

Neither the shaft eccentricity nor bearing vibration differed appreciably from that of normal operation. This evidence indicates that the bearing alignment did not change appreciably during the tests and that the turbine casing has been properly supported to reduce the effects of temperature changes.

After 14 months of service, the exhaust hood was removed for inspection of the low-pressure turbine, and there was no indication of troubles caused by these tests. Only the last 3 stages of the high-pressure turbine could be inspected, and the rubbing strips on the bucket covers had just touched a diaphragm at one spot as had been expected at a time when the throttle steam temperature inadvertently was decreased to an unusually low value while certain feedwater-heater difficulties were being corrected. Time did not permit the removal of the high-pressure-turbine shell so that the inspection of the remainder of the high-pressure turbine was postponed.

Description of Tests

Constant Load. All data at steady-state conditions were measured at 16 test points with loads ranging from 37 to 117 megawatts during the heat-rate tests in the period from February 14 to 22, 1948.

These data are plotted in Fig. 5. For simplicity, only about one third of the temperature measurements are shown. The locations of the thermocouples are shown in Figs. 3 and 4, and the nomenclature and instruments are described in the Appendix.

Starting and Loading. Most starts were made from turning-gear speed after week-end shutdowns with initial turbine temperatures of about 450 F. The total length of starting time, from turning-gear speed to synchronization, was decreased from 98 to 22 min as the test progressed. The loading rate was increased from 0.5 to 2.75 mw per min from no load to a total load of either 40 or 110 mw. In general, the higher loading rates were made with the shorter starts. Ten tests of this nature were made. Data for a typical start and load cycle are shown in Fig. 6.

Throttle steam pressure was held approximately at the rated value of 1250 psig for all starting and loading cycles. For the average start, the throttle steam temperature was near 750 F;

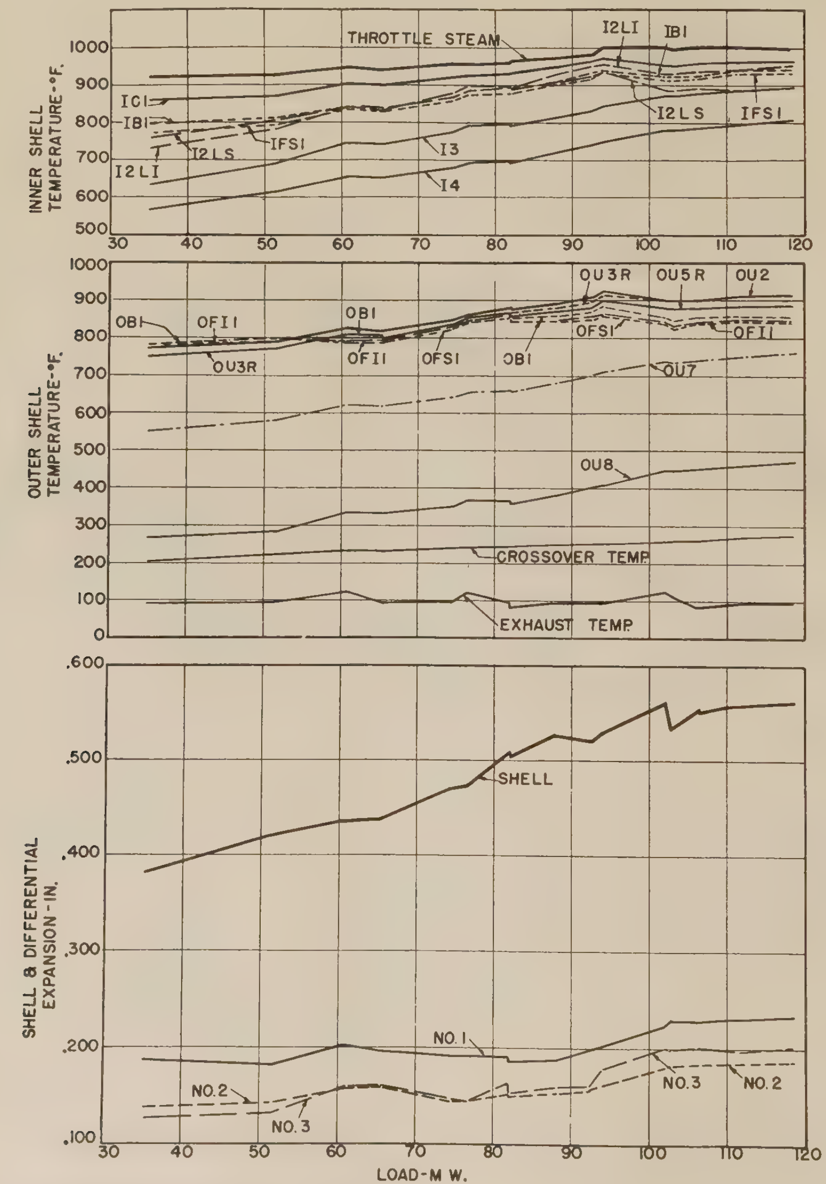


Fig. 5 Temperature and Expansion Data for Steady-State Conditions, Taken During Heat-Rate Tests[7]

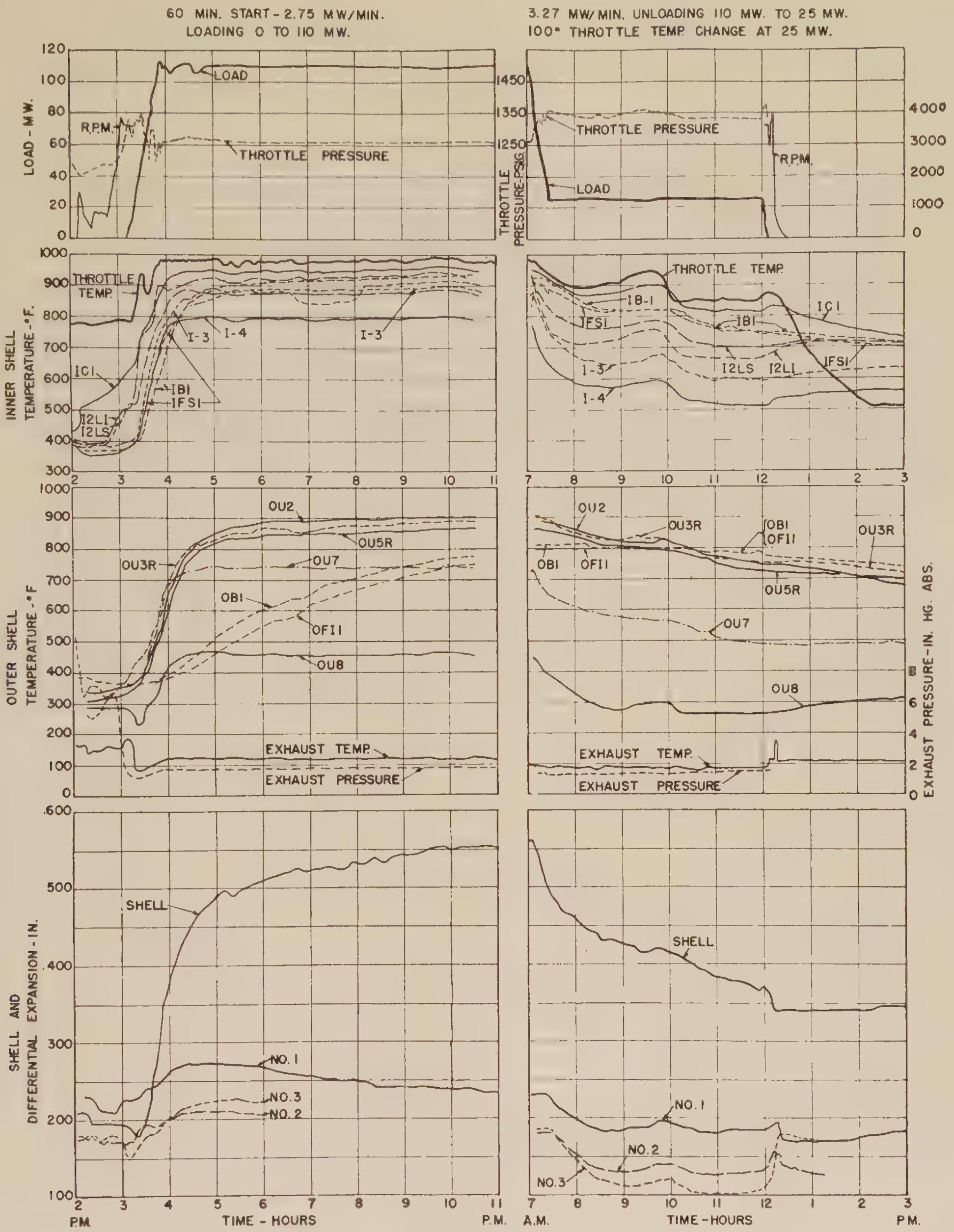


Fig. 6 Temperature and Expansion Data for (1) a 60-Min Start, Followed by Loading to 110 Mw at 2.75 Mw Per Min, and (2) Unloading From 110 Mw to 25 Mw, Followed by a Throttle-Steam-Temperature Drop of 110 F at Rate of 200 F Per Hr and That Followed by a Turbine Shutdown

as soon as loading began it increased rapidly to about 975 F, then dropped rapidly to about 900 F, and then increased as the load increased, reaching 975 or 1000 F at about the same time the load reached 110 mw. The throttle-temperature change shown in Fig. 6 is quite typical, except that the initial peak is low. The exhaust-pressure change is quite typical.

Unloading and Stopping. Unloading rates were varied from 0.5 to 3.27 mw per min from loads of 40 or 110 mw. For some tests, the unloading was stopped at 25 mw, and held until steady-state conditions were established. Eight unloading tests were made.

From the typical unloading-test data in Figs. 6 and 7 it can be shown that the turbine cooling rates are low when the throttle steam temperature is held as high as practical.

With normal operating procedure, the stop valve is closed as soon as the load is off and the machine coasts to a stop in about 20 min. It is immediately put on the turning gear which rotates the shaft at about 3 rpm to prevent bowing because of nonuniform cooling. The stopping cycle introduces no trouble and no tests were planned or made.

Throttle Steam Temperature Change. The throttle steam temperature was changed to 100 F at rates of 50 F and 300 F per hr at a load of 110 mw and at 200 F per hr at a load of 25 mw. Five tests of this type were made. Typical tests are shown in Figs. 6 and 7.

Load Dump and Fast Pickup of Load. The load was dumped by opening the main- and auxiliary-generator circuit breakers. After a no-load period of from 3 to 15 min, loads of over 110 mw were picked up in times of 13 to 19 min. For one test, the auxiliary-generator circuit breaker was not opened. Three tests were made and are shown in Fig. 7.

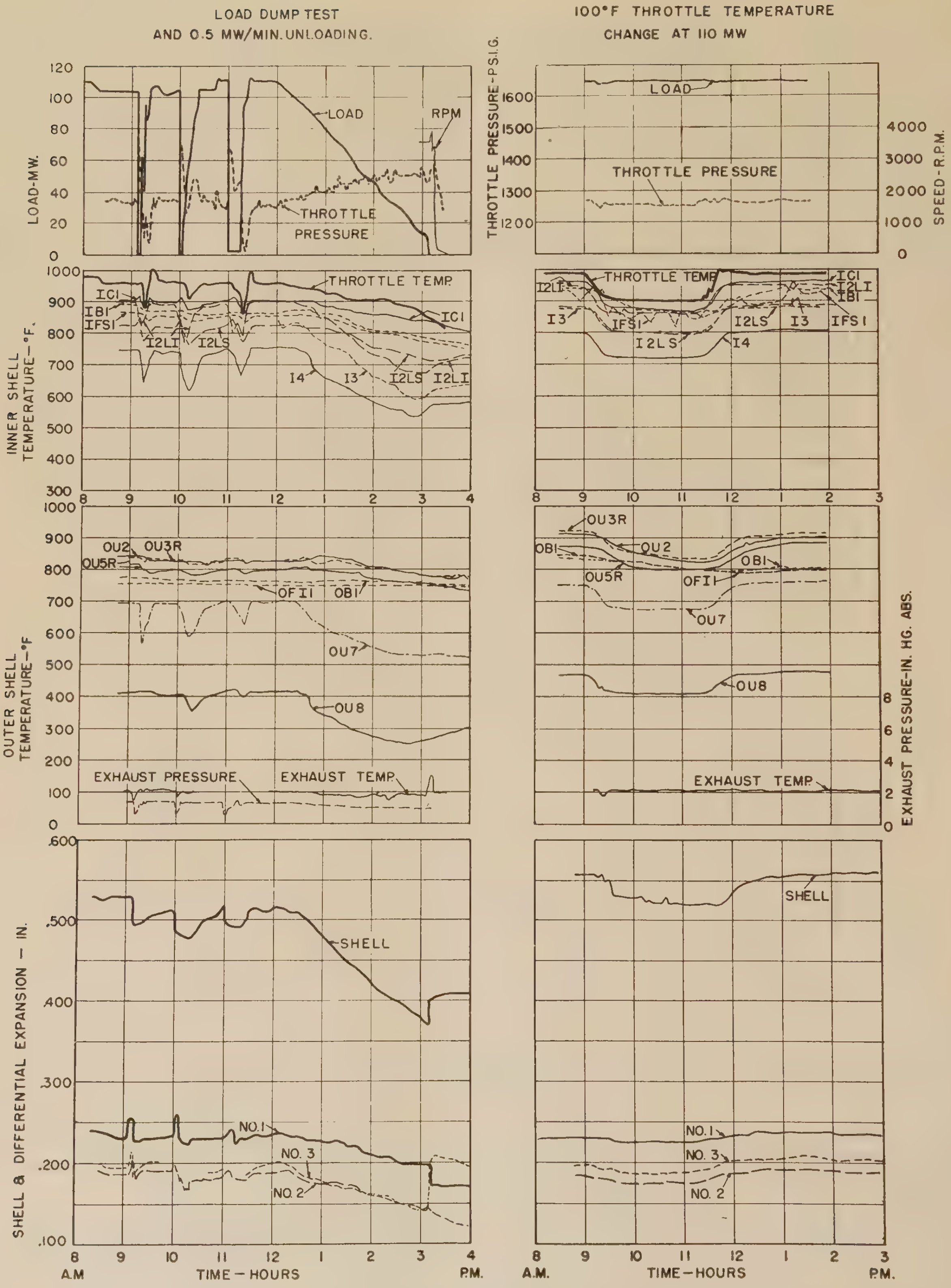


Fig. 7 Temperature and Expansion Data for (1) Three Load Dumps From About 110 Mw, Followed by Rapid Pick-Up of Load, and Unloading From 110 Mw to Shutdown at 0.45 Mw Per Min, and (2) Throttle-Steam-Temperature Change of 100 F at Rate of 300 F Per Hr at a Load of 110 Mw

As a matter of interest, the maximum speed reached by the turbine was 106.1 per cent of normal. A calculated time of only 0.85 sec is required for the turbine generator to reach 110 per cent speed after the circuit breakers are opened with full-load steam flow going through the turbine.

Test Results and Discussion

It would have been desirable to show more of the original data, but their volume made necessary the presentation of these few summarizing curve sheets. As an example of the amount of data obtained, a typical start and load cycle required about 7000 separate observations of temperatures, pressures, and loads, together with the continuous records of load, four expansions, and throttle, crossover, and exhaust temperatures and pressures.

Heating of Turbine. A description of the temperatures during a start is of great assistance in understanding the heating which occurs. Although the internal steam temperatures were not measured, these test data and previous experience indicate that the values quoted are of the right order of magnitude.

With the initial steam temperature at about 750 F during a start, and a calculated temperature drop of about 150 F in the first stage, steam will be supplied to the following turbine stages at a temperature of about 600 F. Steam of this temperature also leaks through the first section of the high-pressure packing and then divides to flow through the remaining sections of the high-pressure packing and between the inner and outer shells into the tenth stage or out to the tenth-stage feedwater heater.

Then, during a start, the valve chest and first-stage nozzle box should approach 750 F, and the remainder of the forward end of the outer and inner shells should approach a temperature of about 550 F. At the tenth stage the steam temperature will drop by perhaps 200 F, so that that section of the inner shell should approach an intermediate temperature between the inner steam of about 400 F and the outer steam of about 600 F. The last 9 stages of the high-pressure turbine will approach temperatures averaging about 350 F. Although the high-pressure turbine tends to equalize in temperature during cooling, the data show that the low-pressure end is usually about 100 F cooler than the high-pressure end. This distribution tends to match metal and steam temperatures. The first stages of the low-pressure turbine should approach a temperature of about 250 F, and the last stages should have temperatures ranging from 100 F to 250 F, depending upon the length of the start.

If the initial temperatures of the high-pressure turbine are approximately 450 F, there should be little change in turbine temperature during a start except in the valve chest and nozzle box. The initial temperature of the low-pressure turbine also should be little changed. Very short starts should then be possible, since the turbine and steam are near the same temperatures. The test data indicate that a starting time of 20 min is satisfactory for this condition.

If the initial temperatures of the high-pressure turbine are approximately 150 F, the start should be somewhat slower, since the turbine is cooler than the steam and heating will take place during the start. The longer start will permit the turbine to approach the 450 F equilibrium temperature for 750 F throttle steam. The test data indicate that a starting time of 50 min is satisfactory for this condition.

If the initial temperature of the high-pressure turbine is 750 F, the turbine should be started with 850 or 900 F steam to prevent cooling toward the 450 F equilibrium temperature which occurs with 750 F throttle steam. If the boiler cannot supply such high-temperature steam at the start, the starting time should be very short, followed by rapid loading to 30 or 40 mw. A starting time of 10 min is suggested for this machine under these conditions.

With the turbine at initial temperatures of about 450 F, some starts show very rapid temperature drops of 50 or 100 F at the steam chest and in sections of the outer shell near the inlet steam pipes. It is believed that more careful heating of the steam pipes and stop valve might prevent this cooling and so keep the subsequent heating gradients at a lower value. Further testing is necessary to establish the proper techniques. A similar effect occurs when the turbine is started with steam colder than the machine and this, likewise, should be avoided.

Typical heating curves at several places in the turbine casing are shown in Figs. 6 and 7. An examination of the rate of heating will give a measure of the stresses caused by the transient. With initial temperatures of the high-pressure turbine of approximately 400 F and, during starts of 20 to 60 min, only the steam chest and two other temperatures on the inner shell show any appreciable change. All other temperatures rise only after loading starts. Fig. 8 shows the maximum rate of heating as a function of the rate of load increase. Not only does the maximum rate of heating vary at different places, but it occurs at different times during the heating cycle. The steam chest shows its highest rate when loading starts and as steam temperature rises very rapidly by about 200 F. The lowest value of the maximum gradient in the high-pressure turbine occurs in the outer-shell flange 10 to 30 min after a load of 110 mw is reached. As an example of heating rates at a particular time, Fig. 9 shows the rates at a load of 50 mw during loading to 110 mw.

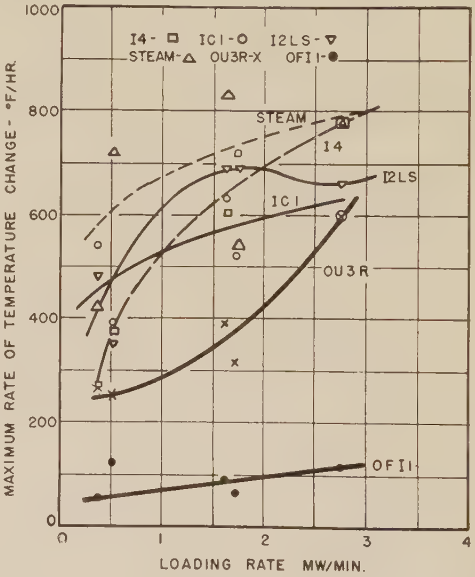


Fig. 8 Maximum Rate of Temperature Change at Representative Locations During Starting and Loading Period

With the initial temperatures of the high-pressure turbine at approximately 150 F, considerable heating takes place before loading. However, the maximum rates are not reached until after the loading starts. The preliminary heating is of such an amount that the subsequent maximum gradients are not greatly different from those that occur when a start is made from an initial turbine temperature of 450 F.

The heating data in Figs. 6 and 8 show that some of the temperature gradients in the turbine are very nearly the same as the temperature gradient of the steam. If the rate of increase of steam temperature were constant during the loading period, the maximum heating gradients in the shell could be reduced to about a quarter or third of the test values.

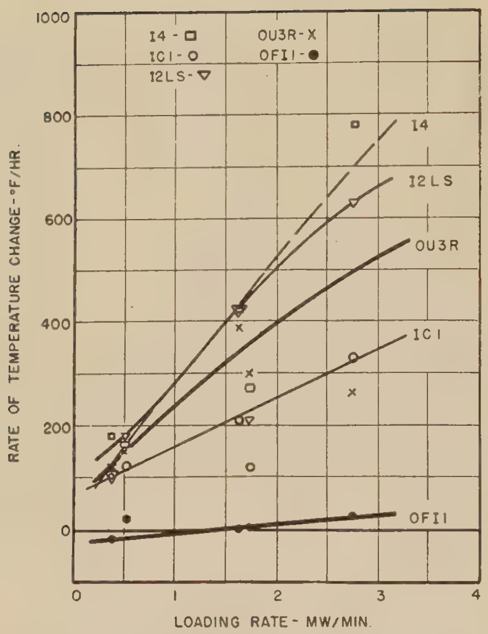


Fig. 9 Rate of Temperature Change at Representative Locations at 50 Mw During Loading From Zero to 110 Mw

A study of the calculated heat-transfer coefficients shows that the rate of turbine heating as the load is increased should not be greatly different from the rate of cooling as the load is decreased for the same change of load and the same rate of load change. The order of difference is probably about 30 per cent for the range of $^1/_3$ to full load. This occurs since, over that load range, the range of temperatures caused by the load change is relatively small. However, when the turbine is started and loaded from room temperature, the temperature differences become relatively large. In stages which change from dry to wet as the load changes, the rate of cooling will be somewhat higher and the general relation given in the foregoing will not apply.

Longitudinal and Circumferential Temperature Distribution. The relation between steam and shell temperatures is shown in Figs. 10 and 11, for loads of 35 and 117 Mw. Of particular interest is the effect of discharging the steam from the first section of the high-pressure packing through the outer shell at the tenth stage. This tends to keep that portion of the outer shell much hotter than the tenth-stage steam. The flanges on both inner and outer shells remain appreciably colder than the corresponding cylindrical sections of the shells. A large section of the inner shell is colder than the outer shell at the light loads. These effects can be explained as a rather complicated combination of the radiation from the steam chest, heat transfer from the packing and valve steam leakage, and heat conduction through the inner shell.

The circumferential temperature distribution was measured at several sections of the outer shell. At constant load, the largest difference around the shell was about 50 F. Differences of about 125 F were observed during both loading and cooling cycles. Previous experience indicated that the differences during cooling would be much greater if the turning gear were not used.

At one shutdown, the exhaust-hood manholes were opened shortly after the machine was shut down and with the turbine still on the turning gear. The ventilation with room air cooled the machine very rapidly. Temperature differences of about 300 F were observed between top and bottom of the outer shell. Certain turbine parts cooled 500 F in 20 min with the major part of the temperature drop at the rate of 3000 F per hr. Such rates are considered to be undesirably high.

Shell and Differential Expansion. Shell expansion is a measure of the average temperature of the shell and, during temperature changes, shows the speed with which the average shell temperature approaches the steady-state value. Differential expansions are important since they indicate the relative movement of the rotor and shell and show when rubbing can occur.

The magnitude of the expansions for steady-state conditions is shown in Fig. 5. (For a more specific description of how these expansions were measured, see the Appendix.) For transient conditions, the greatest change from the initial steady-state values during load and temperature variations, is shown in Fig. 12. With loading rates higher than 1.0 mw per min, the maximum change for the No. 1 differential expansion is practically constant for either loading or unloading and is well below the value, which would cause rubbing. No. 2 and No. 3 reach maximum values of the greatest change and then decrease as loading rate is further increased. Since the clearances in the low-pressure turbine are relatively large, the higher maximum differences cause no concern.

Such deviations of differential expansion are functions not only of loading rate but also of the size of the load change and of the initial temperature of the turbine (or the initial load). The maximum change of the No. 1 differential expansion is plotted in Fig. 13 as a function of loading rate. The dotted and solid lines show the effects of load ranges of 0 to 40 and 0 to 110 mw, respectively. The effect of initial turbine temperature is large and shows why loading rates must be adjusted for this factor.

With the longitudinal variations of shell and steam temperature such as are shown in Figs. 10 and 11, the shell and differential expansions can be calculated for the steady-state conditions and show reasonable checks with the measured values. For this purpose the turbine-rotor temperatures were estimated from the steam and bearing-oil temperatures.

An exact calculation of the differential expansion during transients is difficult because of the irregular shapes, the wide variation of steam velocities and heat-transfer coefficients, and the indeterminate heat-transfer characteristics of metal joints of various types. From the measured differential expansions, estimates of the over-all heating rates of both the rotor and shell can be made. With these rates and the calculated steady-state expansions, maximum values of the differentials during transients can be predicted for any turbine of this type. The rates and predicted expansions check well with previous experience.

Change of Throttle Steam Temperature. A change of throttle steam temperature causes a change of turbine temperatures and expansions. In some load ranges, these changes are much like the effect of a load change at constant throttle temperature.

Since this machine has stage valves which by-pass the second and third stages starting at loads of about 93 mw, the change in temperatures and expansions is small for the load range from 93 to 117 Mw. However, between loads of 37 and 93 Mw, a load change of 33 Mw at constant throttle temperature is somewhat the equivalent of a throttle temperature change of 100 F at constant load. It follows that a load change at the rate of 1.65 Mw per min is similarly the equivalent of a throttle-temperature change at the rate of 300 F per hr under the conditions just given.

The data show that the differential expansions for a load change from 0 to 40 Mw are about the same as those for a throttle-steam-temperature change of 100 F.

This concept of similarity is useful in analyzing and applying the test results, although it should be noted especially that local effects, such as heating of the steam chest and the first-stage nozzle, are very different for the two types of change.

Exhaust-Steam Temperature. During a start, the maximum temperature of the steam in the exhaust hood occurred just be-

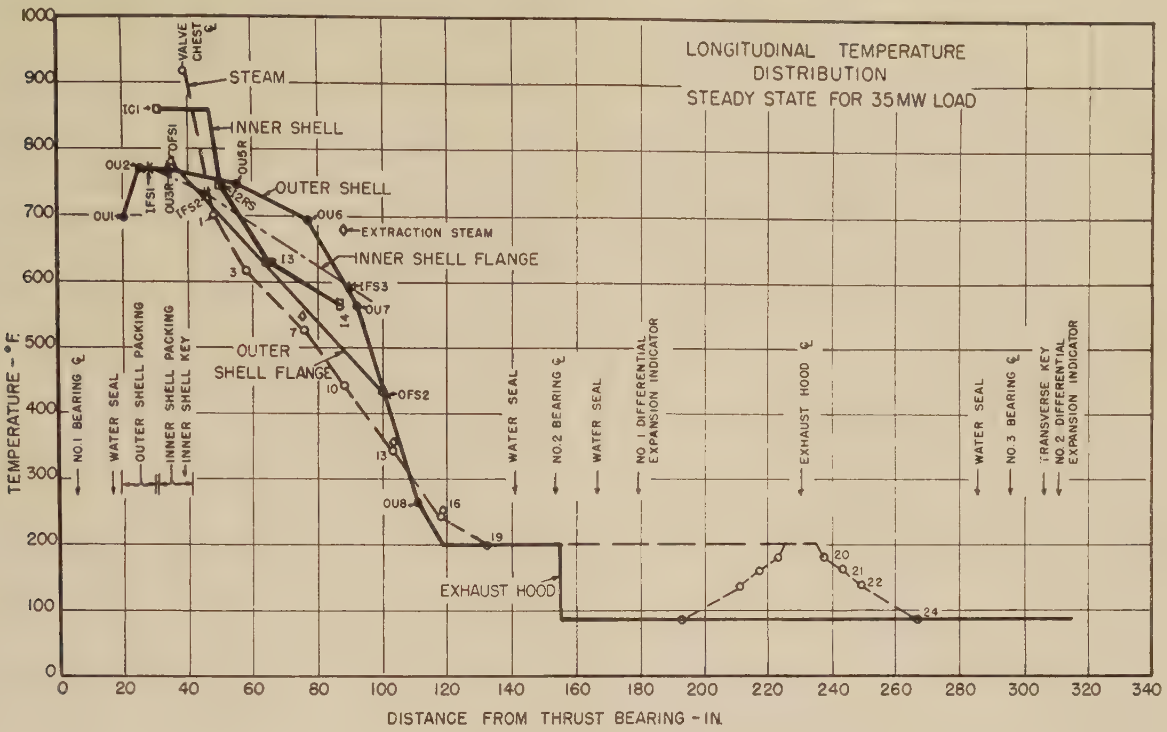


Fig. 10 Longitudinal Temperature Distribution at Constant Load of 35 Mw
(Steam temperatures from heat-rate test.[7])

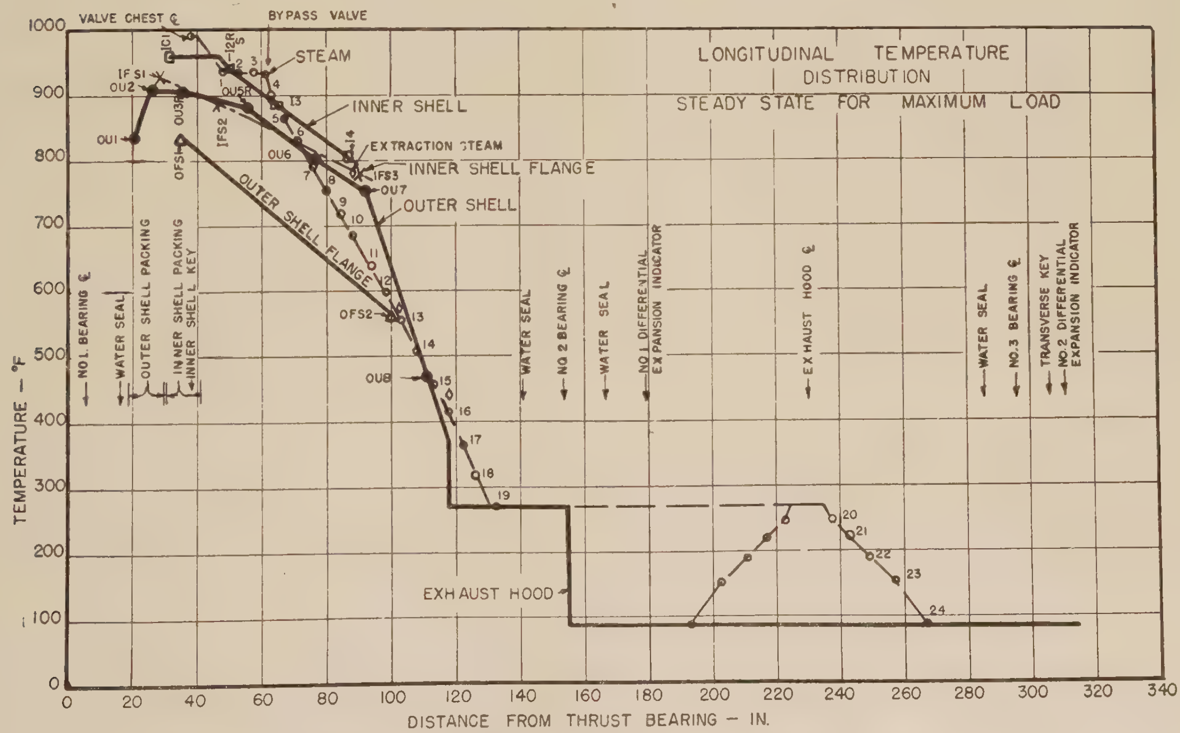


Fig. 11 Longitudinal Temperature Distribution at Constant Load of 117 Mw
(Steam temperatures from heat-rate test.[7])

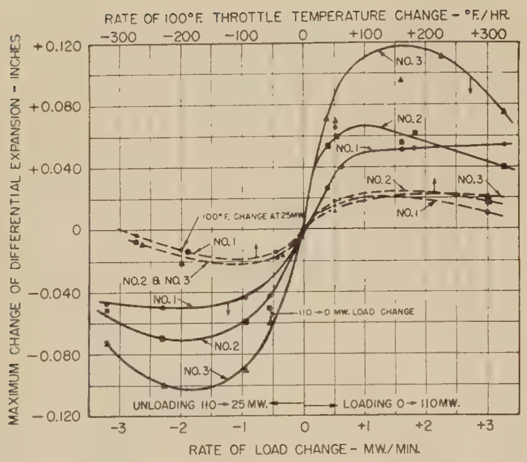


Fig. 12 Maximum Change of Differential Expansion

(Solid lines show maximum change occurring during starting and loading with initial turbine temperature approximately 450 F, and during unloading from full load to 25 mw. Dotted lines show maximum change during 100 F throttle temperature change at constant load.)

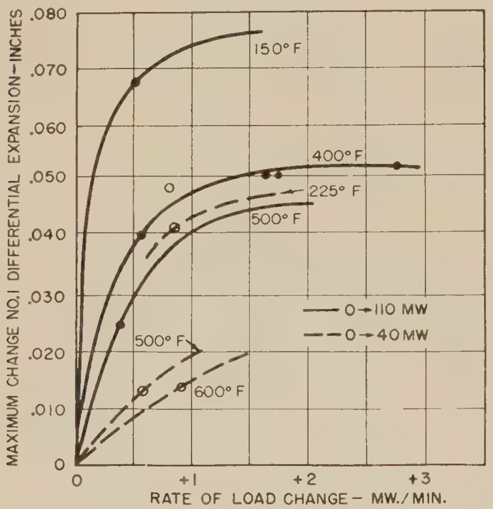


Fig. 13 Maximum Change of Differential Expansion for High-Pressure Turbine Only

(Initial turbine temperature is shown as parameter. Solid lines show load changes from zero to 110 mw. Dotted lines show load changes from zero to 40 mw.)

fore the machine was loaded. The exhaust-steam temperature rises because of the large rotation losses in the last stages. Since those losses are proportional to exhaust pressure, and since that pressure was changing rapidly and was not the same for all tests, the data show some scatter. However, in general, the longer the starting time, the higher the maximum exhaust-steam temperature. The maximum temperature observed during a start was 230 F, which is well below the recommended maximum of 250 F.

At times it is necessary to operate a turbine generator at synchronous speed at no load. Although no tests of this nature were planned, operation for 30 min resulted in a temperature of 225 F. The exhaust pressure at that time was about 1.5 in. Hg. Steady-state conditions had not been reached, and the temperature would have risen further had the test been continued.

Steam-Chest Temperature. The largest observed difference between the throttle steam temperature and the outer surface of the steam chest was 300 F during a starting and loading period. Because of the high temperature drop from steam to metal, the maximum temporary temperature gradient in the metal, calculated from the test data, was 270 F per in. which occurred when loading at 2.75 mw per min.

At steady loads, the gradient would be about 18 F per in., and the stress due to temperature would correspond to about 4500 psi in tension on the outer fibers and about the same amount in compression on the inner fibers. The calculated heat-transfer coefficient on the inside of the steam chest varied from about 3 to 350 Btu/(sq ft) (hr deg F) as the load changed from zero to maximum.

Inner-Shell Temperature. The temperatures of the inner and outer surfaces of the inner shell were measured at two places near the discharge of the first stage. At high loads, the temperature of the inner surface was hotter than the outer, although the steam flowing over each surface was supposed to be at the same temperature. The difference is largely due to the high heat gain on the inside, the high steam velocity, and the large heat loss by radiation from the outer surface. The magnitude of the difference varied from 75 F at constant load to 115 F during loading, and 70 F in the opposite direction during unloading, Fig. 14. The maximum temperature gradient in the inner fibers, as calculated from test data, was 54 F per in., and occurred during loading at the rate of 2.75 mw per min. This is only about twice the average gradient at constant load.

The 75 F temperature difference would cause a stress of about 7300 psi in the cylindrical part of the shell and should cause little concern. The maximum temperature gradient at the inner fibers of the inner shell will cause stresses of about 10,500 psi.

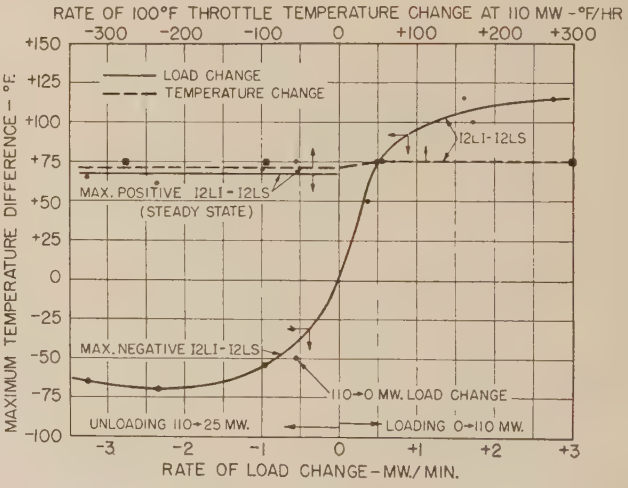
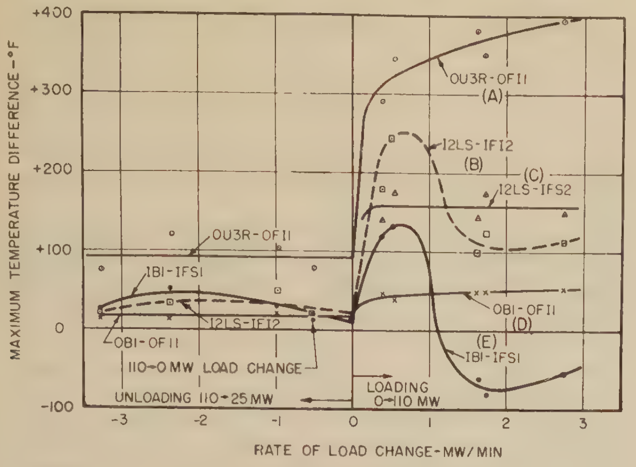


Fig. 14 Maximum Temperature Difference From Inner to Outer Surface of Inner Shell of High-Pressure Turbine at First Stage

(Solid lines show effect of starting and loading, and of unloading cycles. Dotted lines show effect of temperature changes at constant load. During unloading, direction of temperature difference reverses. Both positive, steady-state, and negative maxima are shown.)

Flange and Bolt Temperature. The greatest temperature difference between a bolt and its adjacent flange under steady-state conditions was about 30 F and was about the same for inner-shell studs and outer-shell bolts. During load and temperature changes, the maximum temperature difference between an outer-shell bolt and the adjacent flange was never more than 60 F, Fig. 15(D), for a loading cycle and was smaller for an unloading cycle. The bolt was hotter than the flange, and this indication, to a large degree, may be because the thermocouple inside the flange was not at the bolt center line, Fig. 18(E). Also the radiation from the hot side of the bolthole tends to heat



A Difference between thin cylindrical section of outer shell and interior of adjacent flange.
B Difference between thin cylindrical section of inner shell and interior of adjacent flange.
C Difference between thin cylindrical section of inner shell and outer surface of adjacent flange.
D Difference between outer shell bolt and adjacent flange.
E Difference between inner shell bolt and adjacent flange.

Fig. 15 Maximum Temperature Differences During Load Changes at Representative Locations in Turbine

(Starting and loading data based on initial turbine temperature of about 450 F.)

the bolt faster than the flange, since the small space between bolts greatly restricts the flow of heat to the outside. It should be noted that the temperature of bolt and flange were measured only at one point.

The stress change with a bolt 60 F hotter than the flange would be about 12,000 psi with an incompressible flange and would relieve the initial tension on the bolt. Since the flange is compressed, the difference in stress may be reduced to about 9000 psi.

The temporary temperature differences between the hottest inner-shell bolt and adjacent flange were as large as +125 F and —75 F, Fig. 15E, during a loading period, and the direction of the difference changed with loading rate. The change is believed to be due to the relative amount of heat flow from the interior of the inner shell to the bolt.

Flange and Thin Section of Shell. The largest temperature differences observed in adjacent stressed parts were between the outside of the thin cylindrical section of the shell and the interior of the thick flange directly under the steam pipe (OU3R-OFI1) and occurred during loading. These differences are shown in Fig. 15(A) as a function of loading and unloading rates.

As with the expansions, the important variables which affect this temperature difference are (1) rate of load or throttle-temperature change; (2) magnitude of the change; and (3) initial turbine temperature. Fig. 16 shows the test points with the curves extrapolated to 5 mw per min loading rate for several initial temperatures of the turbine. The extrapolation is believed to be accurate to about 75 F.

If this temperature difference is considered to be a measure of stresses, very interesting inferences can be made from the relation of that difference to loading and unloading rates. If repeated starts with loading at the rate of 1 mw per min can be made safely from an initial temperature of 150 F, with resulting temperature differences of 450 F (from Fig. 16), then loading at the rate of 5 mw per min will result in temperature differences of 530 F, or only 80 F more. Following the same argument, with initial turbine temperatures of 400 F or higher, loading rates of greater than 5 mw per min can be made with no greater probability of trouble than for the start from 150 F and loading rate of 1 mw per min.

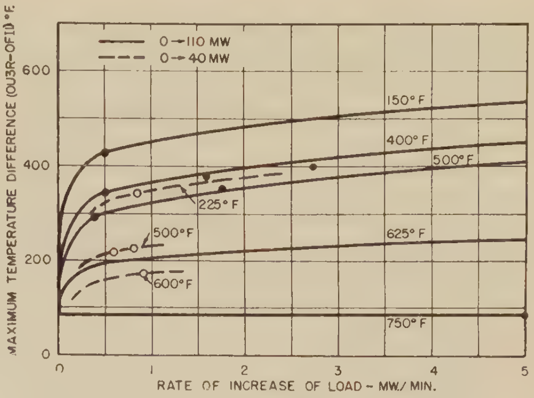


Fig. 16 Maximum Change of Temperature From Thin Cylindrical Section of Outer Shell to Interior of Adjacent Flange

(Initial turbine temperature is shown as parameter. Solid lines are for load change from zero to 110 Mw. Dotted lines are for load change from zero to 40 Mw.)

The temperature differences for load ranges of zero to 40 mw and of zero to 110 mw are shown by the dotted and solid curves in Fig. 16, with initial turbine temperature as the parameter. Initial turbine temperature appears to have a more important effect than the amount of loading when it is considered that the usual steady-state throttle-steam temperature is 80 F lower at a load of 40 mw than 110 mw. This is approximately the difference in temperature between the two sets of curves.

The highest calculated average heat-transfer coefficients on the interior of the outer shell are 3.2 and 11.9 Btu/(sq ft) (hr deg F) for convection and radiation, respectively. The low value of the convection coefficient is due to the low velocity of the steam flowing between the two shells.

The maximum temperature gradient on the inner surface, as calculated from the test data, was about 27 F per in., and was the same at the flange as at the thin section. This would correspond to the negligible stress of about 2000 psi. This stress affects only the inner fibers and is reduced progressively with distance from the inner surface. Other stresses, such as are caused by the difference between the temperature of the thin section of the shell and the adjacent flange, tend to distort the shell. The magnitude of these stresses and the amount of distortion depend upon the configuration and size of the shell and are difficult to analyze.

At steady state, the calculated average temperature gradient is about 5 F per in. in the thin section of the shell and about 3.5 F per in. through the flange. Fig. 17 shows the calculated and measured temperature change of a thin section of the shell and of the flange at the hot end of the outer shell as a function of time after start of loading. The agreement is good and represents a sample of the analysis now progressing.

Other Aspects of Test Results. The limitations of the tests must be realized. No temperatures were measured on the nozzle partitions, diaphragms, or rotor. The bucket and partition edges will heat rapidly because of the small mass and the high heat transfer and may cause permanent and undesirable distortions. The calculated heat-transfer coefficient of the first-stage nozzle partition is about 1000 Btu/(sq ft) (hr deg F).

It is known that the diaphragms will distort when heated or cooled rapidly. However, during the tests there was no evidence that the distortion was sufficient to cause the diaphragm packings to rub the shaft.

Temperature stresses in the shaft and wheels must be con-

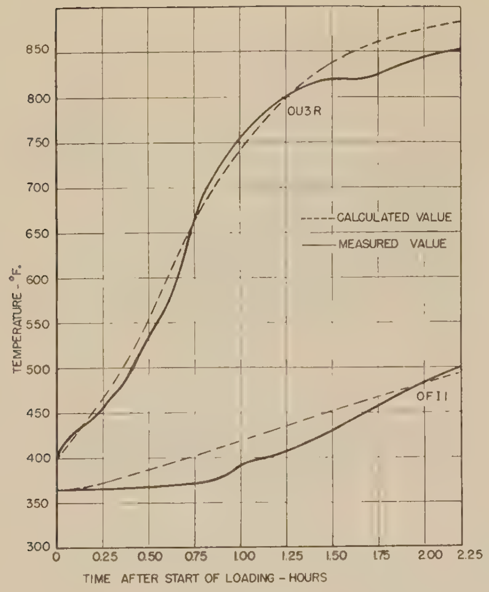


Fig. 17 Measured and Calculated Temperature in Thin Section of Outer Shell and Interior of Adjacent Flange With Load Change From Zero to 110 Mw at Rate of 2.75 Mw Per Min

(Initial temperature of turbine at start about 400 F.)

sidered, and it is hoped that techniques will be developed for measuring temperatures of the rotating parts. Such data will complement the theoretical work which is now proceeding.

Some effects of rapid changes of load or temperature may result in small permanent distortions which may be cumulative and which may be measurable only after many cycles.

Future Test Program

Analysis of these data is continuing and, after further study, it is expected that additional tests will be made at higher rates of load change and several initial turbine temperatures.

A similar 100,000-kw turbine generator but with steam conditions of 1500 psig and 1050 F, and a triple-flow low-pressure section has recently been installed in the Sewaren Generating Station of the Public Service Electric and Gas Company. The machine has been instrumented similarly since more severe steam conditions made temperature and expansion data even more desirable. Although the initial steam temperature is 50 F higher, preliminary data indicate that the outer shell is about 150 F cooler than that of the Essex turbine. This desirable end has been accomplished by an ingenious cooling scheme. It is expected that complete results will be reported later.

Conclusions

Although the analysis is not complete, the following procedures are justified by the test data and are consistent with previous experience. They are recommended specifically for the Unit No. 1 Essex turbine generator.

1 In an emergency, a load of any size can be dropped instantly and, after no-load synchronous-speed operation for as long as 45 min, with $1^1/_2$ in. Hg absolute exhaust pressure, the turbine generator can be loaded as fast as dry steam can be supplied.

2 Conservative starting and loading practice is determined by the rates at which temperatures throughout the machine will approach their steady-state values. Recommended starting times and loading rates are as follows:

Initial turbine temperature, deg F	Starting time, min	Loading rate, mw per min
150	50	1
450	20	3
750	10	5

The rate of throttle-steam-temperature increase should be made as constant as possible to reduce the temperature stress in the shells.

3 Loading or increase of throttle steam temperature can be done safely with respect to both expansion and temperature changes. With the turbine at any steady-state load, load can be increased to any other load at the rate of 5 mw per min at constant throttle temperature, or throttle temperature can be increased by 200 F at the rate of 500 F per hr at constant load.

4 Unloading or decrease of throttle-steam temperature must be considered carefully since the rotor contracts faster than the shell. The cooling gradients are usually low and uniform. As long as dry throttle steam is supplied, any amount of unloading can be done at the rate of 5 mw per min at constant throttle temperature, or throttle temperature can be dropped 200 F at the rate of 500 F per hr at constant load. The safest unloading procedure at any time consists of dropping all load instantly. During any unloading cycle, the throttle temperature should be held as high as possible.

5 The generator starting tests will be reported elsewhere.[5] However, the test results show that there are no restrictions on starting time or loading rate of the generators, although it is recommended in the interests of longer life and lower maintenance that the main-generator field be preheated with a 850-amp current for 20 min immediately prior to operation at speeds above 1200 rpm.

These recommendations for this turbine generator are consistent with those previously published by the turbine manufacturer.[6,7] However, initial turbine temperature is now taken into account and the recommendations are supported by specific test data.

After the remaining tests have been completed and analyzed, it is expected that these recommendations can be changed in the direction of higher rates of loading or temperature change. In interpreting these load rates for comparison to other machines, they should be compared on the basis of per cent of rated load.

The tests show that the Essex Unit No. 1 turbine generator is performing in accordance with design expectations both with respect to operation during transients and economy.[8] Both the manufacturer and the operating company are well satisfied with the performance of the machine and its service record.

Appendix

Thermocouple Nomenclature

The thermocouple locations are shown in Figs. 3 and 4. Designation of each couple has been made so that its approximate location can be deduced. The first letter indicates the inner or outer shell. The second and third letters indicate location on the shell. The number indicates longitudinal position starting at the high-pressure end. The meaning of each letter is given as follows:

---

[6] Discussion by G. B. Warren, Supplement to Trans. AIEE, June, 1942, pp. 398–399.

[7] "Steam Plant Performance on Fluctuating Loads—Part IV—Suitability of Modern Turbine Units," by E. E. Parker and C. W. Elston, Electric Light and Power, Chicago, Ill., April, 1949.

[8] "Heat Rate Tests Results of the 100,000-Kw Essex Turbine Generator," by Vinal S. Renton and Stanford Neal, to be published in this issue of the Transactions, pp. 285–290.

I Inner shell (or inside, if last letter)
O Outer shell
U Upper half
L Lower half (or left side, if last letter)
R Right side
F Flange
B Bolt
C Steam chest

As an example, OU3R would indicate that the couple was in the outer shell, upper half, third position from the high-pressure end, and on the right side.

Instruments and Their Application

The special data required for this test were furnished by 4 expansion indicators and 40 thermocouples. All other data were taken from station instruments. These were the net load, steam pressure and temperature at the throttle, crossover, and exhaust hood, camshaft position, eccentricity of the shaft, vibration at each of the five main bearings, and turbine speed. The exhaust steam temperature was measured with three thermometers in the corners of the hood just above the horizontal joint. The exhaust pressures were measured in the plane of the exhaust-hood discharge.

The expansions were obtained by measuring the change in a magnetic circuit as the distance between the two pertinent parts changed. The instruments were the standard type usually supplied with General Electric Company turbines and were located at the positions shown in Fig. 2. One indicator was installed at the front bearing standard to measure the total expansion of the turbine shell from the exhaust hood to the front bearing standard. The No. 1 differential indicator was mounted at the coupling between the high- and low-pressure turbines and measured the difference in high-pressure-turbine rotor and shell expansion. That difference was measured from the thrust bearing which is located in the front bearing standard. The No. 2 differential indicator was mounted near the coupling between the turbine and the generator and measured the difference in expansion of the entire turbine rotor and shell from thrust bearing to generator coupling. The No. 3 differential indicator was mounted at the coupling between the main and house generators and measured the movement of the turbine and generator shafts with respect to the foundation.

The difference between the indications of No. 1 and No. 2 is the difference in expansion of the low-pressure-turbine rotor and the exhaust hood. Since both the No. 2 and No. 3 indicators are essentially mounted on the foundation, the difference between their indications is the thermal expansion of the main-generator rotor less its elastic contraction in the longitudinal direction at speed relative to the foundation. When the expansion of the foundation is added to that difference, it is possible to deduce the average temperature of the rotor with a good degree of accuracy. The maximum thermal change in length of the generator foundation was small and was about 0.018 in. between room temperature and that corresponding to steady state, full load.

The No. 1 expansion indicator was in the exhaust hood in a steam atmosphere which occasionally reached 230 F, but no trouble was experienced for that reason. The other three indicators were in air at room temperature. The accuracy of the expansion measurement is believed to be from 0.002 to 0.004 in. No. 1 and the shell indicators were calibrated in place. No. 2 and No. 3 were calibrated before installation.

The shell temperatures were measured with 40 chromel-alumel thermocouples which were covered with silicone-varnish-impregnated glass insulation and were 80 ft long. The high-temperature ends of the couples were enclosed in stainless-steel tubing, and the junction of each couple was brazed to the tube, making a steamtight joint as shown in Fig. 18(A). The method of attachment of the couple to the shell is shown in Fig. 18(B). Fig. 18(C) shows the steamtight seal at the outer shell. The seal operated between steam conditions of 790 F, 400 psia and atmosphere. The steel sheath of each couple was brazed individually to the sealing diaphragm. The connecting pipe was made long enough

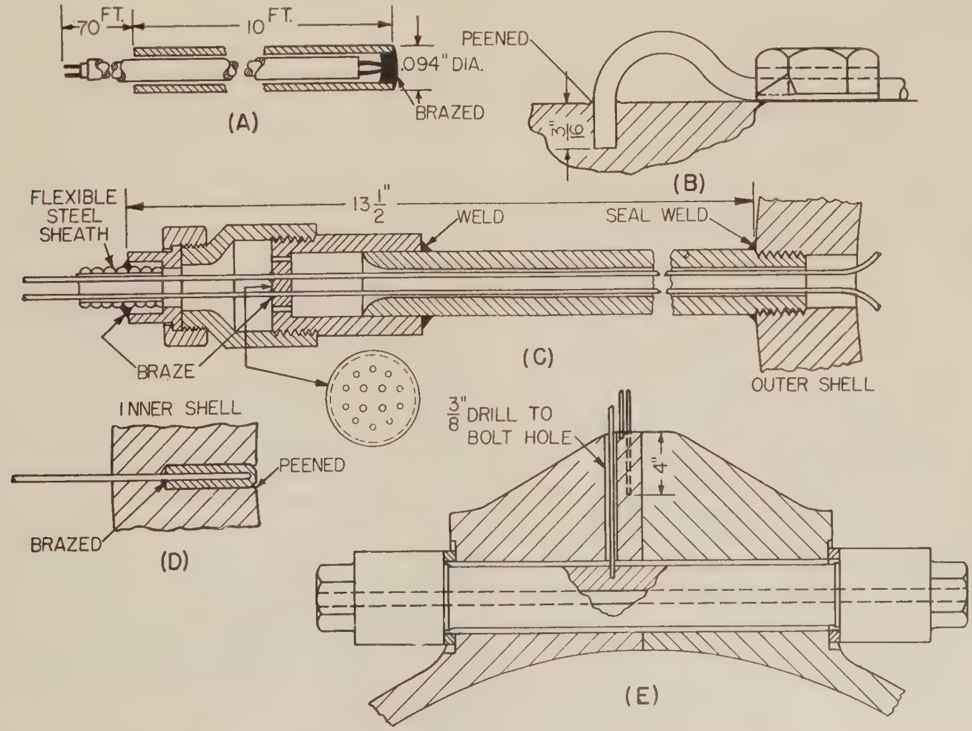


Fig. 18 Details of Thermocouple Construction and Installation

to project through the turbine lagging so that the brazed assembly connections could be inspected or serviced. For protection of the fragile insulation, the couples were enclosed in a flexible steel sheath from the connection at the turbine to the potentiometer terminal board.

It was considered desirable to measure the temperature of the inner and outer surfaces of the inner shell at the first stage to determine the stresses due to unequal temperatures. The method of measuring the inner surface temperature is shown in Fig. 18(D). The couple was inserted through the hole in the shell and brazed to the cylindrical plug. The assembly was then pulled back into the shell, and the plug was peened in place. This construction was rugged and has given no trouble. The bolt and flange couples are shown in Fig. 18(E). All thermocouple potentials were measured by a self-balancing direct-reading potentiometer, which printed the temperatures at 4-sec intervals.

The couples were calibrated before installation, and the potentiometer was frequently checked during the progress of the test. It was expected that the thermocouples and potentiometer have a combined error of less than 4 F.

Some troubles were experienced during the 14 months of operation, and four of the couples are now open-circuited. Based upon this experience, it is believed that a couple can be made which will give accurate temperature readings for several years.

Acknowledgment

The authors express their appreciation to Messrs. E. M. Kratz and J. E. Phillips who were largely responsible for the instrumentation, testing, and evaluation of the data, and Messrs. H. G. Burk and J. Allhusen who supervised the operation of the unit during the tests.

Discussion

J. C. Falkner.[9] The tests on the Essex machine have confirmed the Consolidated Edison Company's tests on quick starting. That loading rates from 3.5 to 5 mw per min are safe is one of the most important conclusions of these tests, as many times we are faced with the necessity of loading machines rapidly to take care of sudden dark clouds, or thunderstorms. As the temperature of our high-pressure units after a 6- to 8-hr shutdown will be at 700 to 800 F, the rate of loading as shown by the paper should be as fast as desirable.

The Consolidated Edison Company of New York, like many other operating companies, had for years followed operating instructions given by the manufacturers as to how fast a turbine should be started, loaded, and what rate of steam-temperature change could be made safely during this starting and loading. In April, 1946, the company, pressed by steam conditions, departed from the manufacturers' instructions and began quick starting of high-pressure topping units. Since that time over 200 quick starts have been made, and there has not been a single instance where trouble developed during a start. There is every indication that the method is much better than that originally given by the manufacturers. However, after quick starting, a limitation of 100 F per hr rate of temperature change was maintained during loading. Loading rates have been held to values of $^1/_2$ to 1 mw per min after an initial loading of 4 mw. If the initial temperature of the turbine casing is 150 F or more lower than the steam temperature, the start is made at a slower rate. An hour is usually allowed for a start under these conditions.

The Consolidated Edison Company opened one of the topping turbines at Waterside after operating 46,000 hr between November, 1942, and August, 1948. During this time there were 32 quick starts of approximately 15 min each, 74 water washes to wash off the phosphates, and four caustic washes to remove silica scale from the buckets. The machine was found in excellent condition, thus proving beyond any doubt that quick starting had not been injurious to the mechanical structure of the unit.

The quick starting of the turbines at Waterside led to trials at Sherman Creek Station, but conditions there were different as there is only one boiler per unit, while at Waterside there are two per unit with only one of the two boilers being used for starting. Therefore, temperatures as high as 700–750 F can easily be had on the starts at Waterside. On the 1,000,000 lb of steam per hr boiler at Sherman Creek, however, the steam temperature could not be brought over 500 F until a few minutes after starting. This meant that for a 6- to 8-hr shutdown the turbine metal temperature had to be reduced deliberately just before shutdown so that the initial steam temperature on the start would not cause a sudden drop and then a quick rise in cylinder temperature. In an effort to increase this initial steam temperature four oil burners were installed high up in the furnace and pointed directly at the superheater. The results of typical quick starts with and without these oil burners are shown in Fig. 19 herewith.

In the start with oil burners the initial steam-chest temperature was allowed to be much higher than had been the usual practice, or 720 F, and the rolling time was cut down to 10 min. The turbine metal temperature drop was 50 F from 720 F to 670 F, and the rise was 100 F in 23 min, and only 150 F in the first hour. However, the steam temperature at the start was still only about 20 F above saturation, despite the use of the oil burners, and this temperature increased rapidly only after the throttle was opened, rising 105 F in the first 5 min. At the present time an additional by-pass around the topping turbine to the low-pressure machines is being designed so that the superheater will be cleared of water and the desired steam temperature will be obtained before opening the throttle. When this is installed the problem of steam-temperature control during starting will have been solved.

One of the latest and most important supervisory instruments used on turbine generators is the differential-expansion recorder which shows continuously the relative axial position of the cylinder and spindle and warns the operator of danger from an axial rub due to the spindle shrinking too fast in reference to the cylinder. In the paper the authors demonstrate that the differential-expansion recorder shows very little differential motion between spindle and cylinder when the turbine shell metal temperatures are between 400 and 450 F and the steam being admitted is at 700 to 750 F. This small relative motion is probably due to the fact that the heat in the steam is being dissipated and its temperature degraded while passing through the throttle and governor valve, and steam-chest structure.

Consolidated Edison Company, after the success of quick starting of topping turbines, tried the method on a 375-psi, 700 F, 160,000-kw tandem-compound condensing unit, and also on a 185-psi, 500 F, single-cylinder, 60,000-kw condensing unit. These trials were as successful as with the topping units, starting time being lowered on the large machines from $1^1/_2$ hr to 20 min, and on the smaller machines from $1^1/_2$ hr to 15 min. A decided improvement was obtained at the exhaust end of both units by lowering exhaust end temperatures immediately after the last-stage wheels by as much as 75 F to 100 F.

The Consolidated Edison Company expects to apply the quick-starting technique to all topping and condensing units on the system, and thereby realize an estimated annual saving of $250,000.

In addition to the operating saving just mentioned, there are

---

[9] Manager, Electric Production Department, Consolidated Edison Company, New York, N. Y.

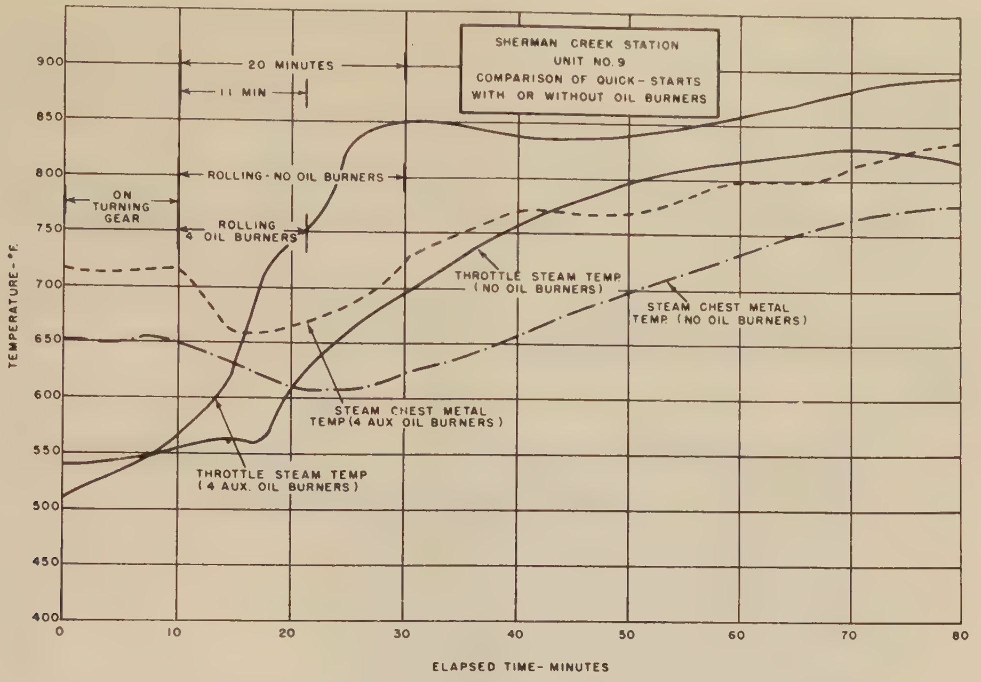


Fig. 19 Results of Quick Starts With and Without Auxiliary Oil Burners

many practical operating advantages which result from quick starting. A notable instance occurred on August 27, 1948, when the Manhattan Cooper Square network was de-energized because of failures of generator cables in Waterside No. 2, causing a simultaneous outage of three out of the four topping units in this station. As soon as the first generator cables were temporarily repaired on Unit No. 4, this machine was brought up to speed and was generating power in less than 15 min and the network was re-energized. If the old original 3-hr start had still been in vogue at the time, the network would have remained out of service at least $2^1/_2$ hr longer.

The short starting time relieves the man-power problem in the turbine room of each station during the start-up period. With the quick-starting technique perfected it is possible to put machines in service one after the other with fewer men, who can concentrate on one machine instead of on several.

With the potentialities of quick starting of large high-pressure turbine generators fully realized, the Consolidated Edison Company is now focusing its thoughts on quick starting of high-pressure boilers after week-end shutdowns. We think that it will be possible to start such boilers from cold and have them up to full pressure in 1 hr or less. Test work is being planned at present to develop the details of such high-pressure-boiler quick-start procedure and, if expectations are realized, it is felt that the economical advantages of the integrated quick starting of boilers and turbines will have been fully exploited. However, we believe that fast shutdown and quick starting of high-temperature high-pressure boilers will be best accomplished when an all-welded construction is used to join waterwall and generating tubes to their headers and drums.

H. H. Poor.[10] To obtain information on boiler performance during rapid load swings, and to investigate the permissible rate of load pickup on a hot boiler, the writer's company participated in the last three load-dump and quick-pickup tests in the program at Essex Station described in the paper.

Fig. 20 of this discussion shows a sectional side elevation of the steam-generating unit, which is a B&W radiant boiler designed to deliver 900,000 lb per hr of steam continuously at 1350 psi and 985 F from feedwater at 455 F, with a 4-hr peak load of 1,000,000 lb per hr. Twelve circular burners, arranged for either pulverized-coal or oil firing, deliver fuel to the slag-tap furnace. The burner region of the furnace is constructed of close-spaced tubes fully studded and covered with plastic chrome ore; the remaining upper portion of the furnace is of bare tube-to-tube construction. The furnace is divided by a central water-cooled division wall to provide more cooling surface in the furnace, without increasing furnace volume, than would be possible without such a wall. There is a boiler screen of three half-rows and four full rows of boiler tubes, two-stage pendant superheater with interstage spray attemperation, economizer, and air heater.

During the tests oil was fired because of greater operating flexibility, as compared with pulverized-coal firing. The boiler master controller, feedwater flow rate, fuel-oil flow rate, induced-draft fans, and forced-draft fans were on hand control; the spray attemperators were on full-automatic control. The feed pumps were on automatic control, but during load pickup were adjusted manually. Operators were stationed at the burners in order to remove the atomizers from service or put them back in service as rapidly as possible.

The last test was the most interesting because of the very rapid increase in load after 15 min at no load, following the full-load trip-out. As shown in Fig. 21 herewith, the trip-out took place at 11:00 a.m. when the boiler was delivering 970,000 lb per hr. The steam flow dropped immediately to the quantity necessary to roll the turbine at rated speed under no load (this was a reading of 70,000 lb per hr on the steam-flow chart). After $14^1/_2$ min, load pickup was started by an initial load increase of about 50,000 kw, and a steam flow of 920,000 lb per hr was regained in $3^1/_2$ min.

The drum pressure just before the trip-out was 1400 psi. When the steam flow was cut off, the superheater-outlet pressure rose at once and the lowest set valve popped about 1 sec after the trip-out. All three superheater-outlet safety valves and some or

---

[10] Staff Engineer, The Babcock & Wilcox Company, New York, N. Y. Mem. ASME.

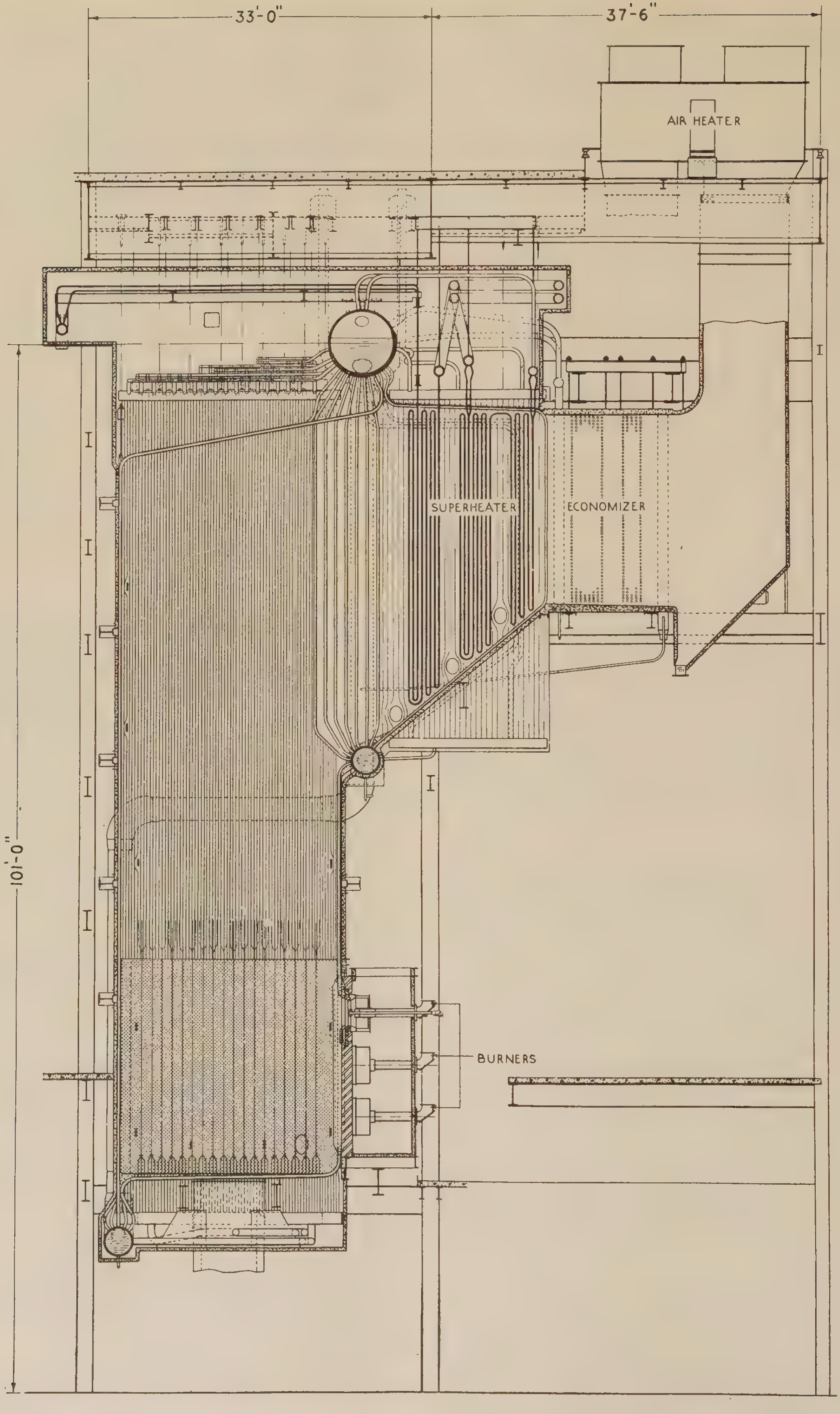


Fig. 20 Babcock & Wilcox Radiant Boiler, Essex Generating Station, Public Service Electric & Gas Company, Newark, N. J.

all of the drum safety valves opened, relieving the pressure as indicated in Fig. 21. The last valve closed a little more than 7 min after trip-out. The heat supplied by the two oil lighters remaining in service was sufficient to raise the steam pressure so that the low set valve reopened 10 min after trip-out at a pressure of 1370 psi, and reseated $2^{1}/_{2}$ min later at 1340 psi. When the load was applied, the drum pressure dropped to a minimum of 1230 psi, leveled out at 1250 psi for some time, and then climbed back to normal operating range.

Water level was measured by the panel-board recorder and was checked during the time that the water was visible in the gage glass by an observer stationed at the glass. At trip-out the water level dropped from $+^{1}/_{2}$ in. to —13 in., but had returned to the bottom of the gage glass 6 in. below the drum center line at the end of 1 min. The water level was held at 5 in. below the drum center line when the load pickup was approached. When the first block of load (approximately 50,000 kw) was applied to the turbine and the steam flow jumped from practically zero to 550,000 lb per hr, the water level rose 14 in. to a level 9 in. above the drum center line. The load was held at this value for about 2 min, during which time the drum level came back to —4 in., remaining in sight in the gage glass, except momentarily, during the remainder of the load pickup.

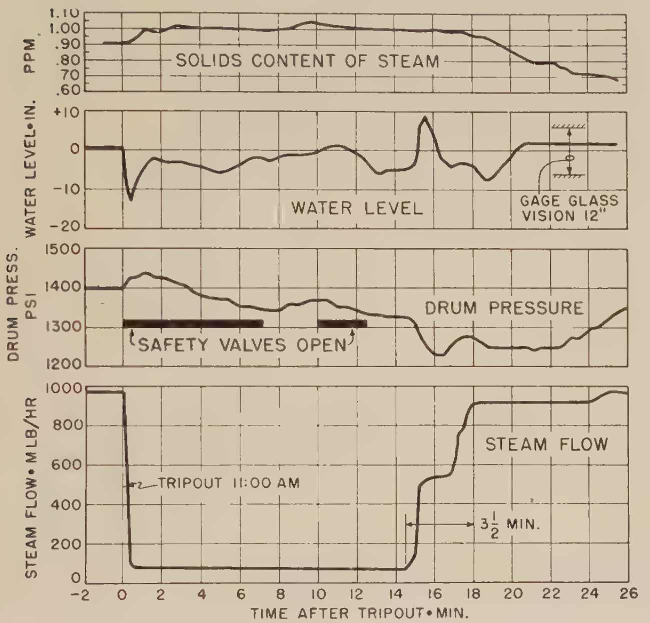


Fig. 21 Records Taken During Trip-Out Test

As a check for carry-over, saturated steam was taken through a sampling nozzle located in the center saturated-steam connection from the top of the drum, and passed through a cooling coil directly into a conductivity cell, with no attempt to remove dissolved gases. Before the trip-out the conductivity-cell indication corresponded to a solids content of 0.9 ppm. After the trip-out the indication increased to 1.0 ppm and finally to slightly above 1.0 ppm. There was no sign of carry-over increase when the load was picked up, despite the rise in water level, and, after full-load steam flow had been reached, the indicated solids content gradually decreased to less than 0.7 ppm.

Gas temperature entering the superheater was measured by a high-velocity thermocouple located in the gas stream halfway between the roof and floor at the superheater inlet, 10 ft from the right side wall. The gas temperature dropped from 2100 F before the trip-out to 1100 F in 1 min, and then drifted down in irregular fashion to 800 F during the low-load period. When load was resumed the gas temperature rose again to 2100 F.

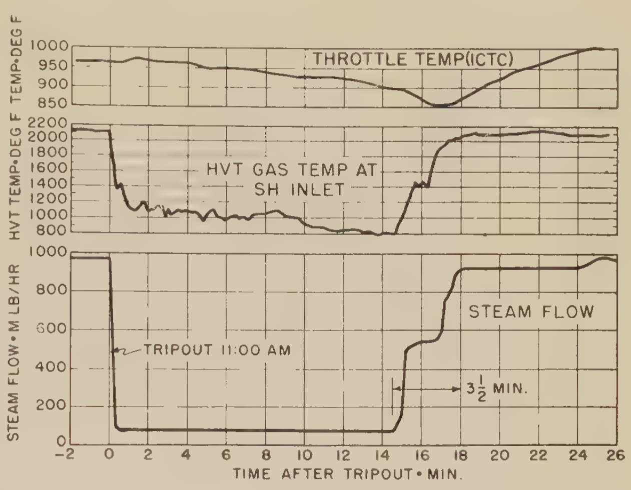


Fig. 22 Record of Conditions During and After Trip-Out

Steam temperature at the turbine throttle was approximated by a bare 22-gage iron-constantan thermocouple inserted in an idle thermometer well at the turbine inlet. Due to the thermal inertia of the thermometer-well assembly, the temperature reading did not follow the steam temperature very closely. The steam temperature indication before the trip-out was 960 F, and during the no-load period slowly drifted down to 900 F. During the load pickup, a large quantity of saturated steam was passed through the superheater before the normal superheater absorption rate had been re-established, with the result that the steam temperature indication dropped to 850 F during load pickup. Then, as superheater absorption returned to normal the steam temperature climbed to 1000 F.

In so far as the boiler proper was concerned the limiting factor with respect to rate of load pickup was water level in the drum. Under the condition of test the rate of load increase was approximately 240,000 lb per hr per min; as a result of these tests we conclude that with optimum regulation and careful co-ordination of the feedwater with the rate of load increase, it would be possible to bring this boiler from no load to full load in somewhat less than 3 min, at a rate in excess of 300,000 lb per hr per min, without raising the water level more than 10 in. above the drum center line, and without increasing the solids content of the steam.

Authors' Closure

The authors wish to emphasize again that the rates of loading of 5 mw per min should be considered as applying only to this size machine and should be expressed as a percentage of rated machine capacity when these results are interpreted as applying to machines of different construction and capacity.

Mr. Falkner's use of burners located high in the boiler to give high steam temperatures at starting is a very interesting and apparently necessary development and should make safer and more practical the starting of a turbine after a relatively short shutdown.

One means of reducing the thermal stresses during fast loading is by holding the temperature of the steam high while the unit is operating in spinning reserve. Since most boilers have a drooping characteristic, the same high location of burners may be desirable when a turbine is held as spinning reserve. It is understood that some West Coast power companies are investigating this procedure. It is to be understood that the higher throttle temperatures at low loads may require slightly higher minimum loads to maintain low exhaust-hood temperatures.

The annual saving of $250,000, as quoted by Mr. Falkner, is impressive enough to justify the large amount of experiment necessary to establish the optimum starting and loading rates. The discussions of both Mr. Falkner and Mr. Poor illustrate the necessity of considering starting and loading the entire power plant, rather than the turbine, boiler, or auxiliaries alone. Mr. Poor's discussion also indicates a direction for boiler design if fast pickup of load is made a requirement.

A study of the temperature data taken by Mr. Falkner and others shows that quick starting of a hot turbine should not impose undesirable stresses on the machine when the quick starting is made under proper conditions and supervision. The inspection of the Consolidated Edison Company turbines, showing that a large number of quick starts had been made with no damage or increased maintenance, confirms this study. However, quick starting is by no means a universal panacea for all starting or loading difficulties. There is no substitute for close mechanical supervision of any piece of machinery during such times. If anything, supervision during rapid starting and loading is even more important than at other times.

A primary requirement in starting and loading is that thermal stresses should be held to allowable limits. This involves the consideration of steam and metal temperatures and the heat-transfer coefficients. After the allowable limits for different material and different type of construction have been established by experience over a number of years, it will then be possible to establish starting times and loading rates for the allowable limits of thermal stress. It is probable that the loading rates will not be uniform over the complete load range of the machine and will be affected by the initial conditions of throttle steam and metal temperatures, as well as the conditions of starting. With further tests made possible by the co-operation of the operators and the manufacturers, further knowledge of this subject will be obtained and power-generating equipment may be operated nearer the point of its greatest usefulness.

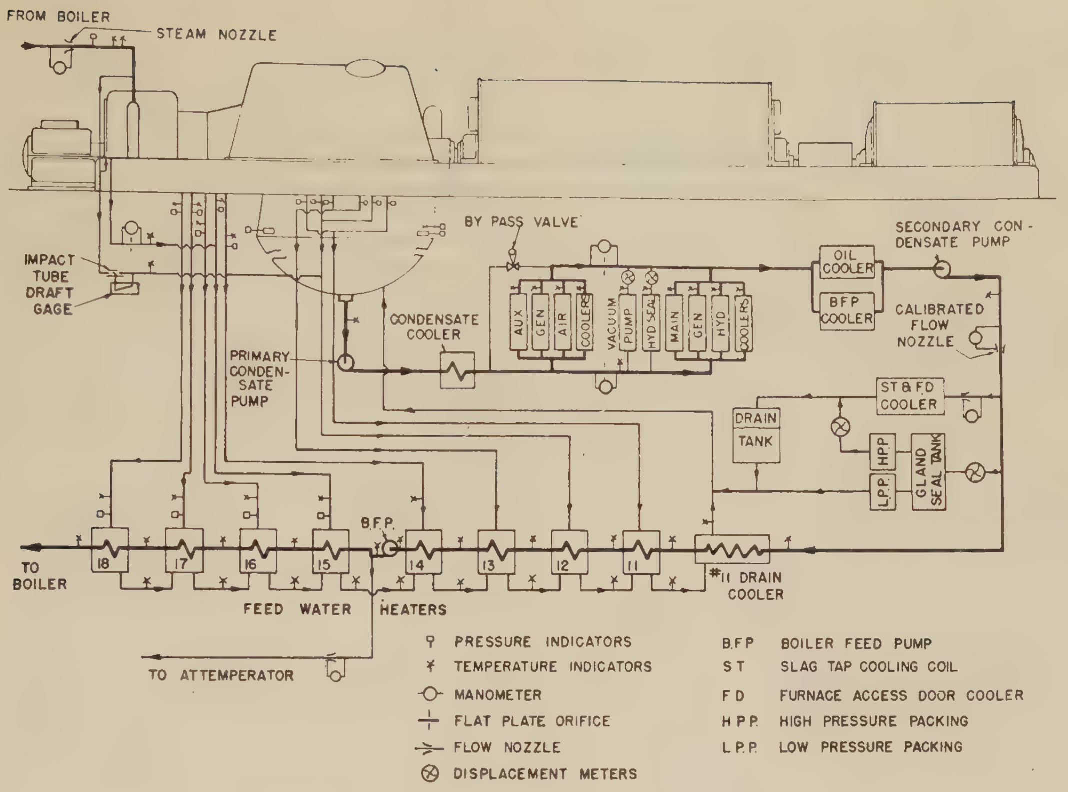


Fig. 1 Diagram of Cycle and Location of Instruments

Heat-Rate Test Results of the 100,000-Kw Essex Turbine Generator

By VINAL S. RENTON[1] and STANFORD NEAL[2]

A 3600-rpm, tandem compound, 100,000-kw turbine generator was recently installed at the Essex Generating Station of the Public Service Electric and Gas Company of New Jersey. Since it was, at that time, the largest machine of its type ever built, the first of a new class, and had initial steam conditions of 1250 psig and 1000 F, careful and extensive heat-rate tests were made jointly by the operating company and the manufacturer. The paper presents the results of the tests as an example of progress made in power-generating equipment.

Introduction

On December 18, 1947, the largest 3600-rpm, tandem-compound steam turbine-generator ever built up to that time was placed in service as No. 1 unit at the Essex Generating Station of the Public Service Electric and Gas Company, Newark, N. J. Since it was the first of a new class of units for the manufacturer and the operating company, extensive tests were made to determine its heat rate and operating characteristics. The operating characteristics are reported in a separate paper.[3]

The heat-rate tests, with which this paper deals, were made in the period from February 14 to February 24, 1948, jointly by the Testing Laboratory of the Public Service Electric and Gas Company and the Turbine Engineering Division of the General Electric Company.

Results of the test were consistent with and slightly better than the guarantees. They indicate that this machine, of advanced design performs in accordance with anticipations.

Cycle and Turbine Design

The turbine generator with its unit boiler, condenser, and auxiliaries, was designed to fit into the space occupied by the existing No. 1 unit, rated at 20 mw, which had been installed in 1915. The specifications for the new No. 1 unit called for a highly efficient cycle which was to have its best heat rate at loads between 70 and 90 mw.

The feedwater-heating arrangement is shown in Fig. 1. It is believed that this is the first time in this country that as many

---

[1] Engineer, Electric Engineering Department, Public Service Electric and Gas Company, Newark, N. J. Mem. ASME.

[2] Section Engineer, Steam Research Section, Steam Turbine Engineering Division, General Electric Company, Schenectady, N. Y. Mem. ASME.

Contributed by the Power Division and presented at the Semi-Annual Meeting, San Francisco, Calif., June 27–30, 1949, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 49—SA-19.

[3] "Operating Characteristics of the 100,000-Kw Essex Turbine Generator," by Stanford Neal and Vinal S. Renton, published in this issue of the Transactions, pages 267–284.

as eight stages of feedwater heating have been used for one turbine. The heater system is also unusual in that the drains are cascaded, then passed through a drain cooler and to the hot well, where deaeration is accomplished.

As shown in Fig. 1, most of the losses are used to heat condensate. For conditions of high turbine exhaust pressure with the resulting high temperature of the condensate, a cooler is provided to reduce the condensate temperature for cooling the auxiliaries.

Description of the Turbine Generator

The semisection of the turbine is shown in Fig. 2. A mechanical description is reported elsewhere,[4] and the following discussion is limited to points pertinent to a study of the heat-rate results.

The tandem-compound turbine has a nominal rating of 100,000 kw, and the steam conditions are 1250 psig 1000 F. The high-pressure section has 19 stages with extraction connections from the 7th, 10th, 13th, 16th, and 19th stages. The double-flow low-pressure section is on the same shaft and has 5 stages with extraction connections from each of the two 20th, 21st, and 22nd stages.

The three feedwater heaters supplied, with steam from the three latter extractions, are placed horizontally in the condenser neck and directly under the low-pressure turbine to conserve space and reduce the length of the extraction piping. Model tests were made jointly by the manufacturers of the turbine and condenser to determine the best location of the heaters. The 23rd and 24th-stage moisture catchers are drained to the hot well.

Steam flow and load are controlled by eight primary valves which admit steam to the first stage, and four stage valves which by-pass steam from the discharge of the 1st stage to the inlet of the 4th stage. The valving sequence is arranged so that seven of the primary valves open consecutively for the first seven admissions. The four stage valves are next opened together for the 8th admission, and the eighth primary valve is opened for the 9th admission.

The shaft of the high-pressure turbine is sealed by a labyrinth packing and an impeller-type water seal. The labyrinth packing at the high-pressure end has three sections with leakage steam from the 1st section going through the space between the inner and outer turbine shells and out to the 10th-stage heater. The leakage steam from the second section is piped to the 16th-stage heater, and that from the third section to the 21st-stage heater. The labyrinth packing at the low-pressure end has only one section, which is piped to the 21st-stage heater. The shaft of the low-pressure turbine has only impeller-type water seals.

The main generator is rated 111,765 kva at 0.85 power factor, with 15 psig hydrogen pressure, and 117,647 kva at 0.85 power factor with 25 psig hydrogen pressure. The house or auxiliary generator is air cooled and rated at 7500 kw at 0.80 power factor. Separate motor-driven exciters supply the field electrical requirements.

The turbine and the two generators are coupled solidly to one shaft, and the unit has an over-all length of 77 ft, $7^{1}/_{2}$ in. and weighs 1,120,000 lb.

Moisture Removal

The quantity of drain flow was calculated by heat balances around each heater individually. Since all of the heater drains were cascaded through a drain cooler to the condenser, the quantity of total drain flow also was calculated by a heat balance at the drain cooler. The drain flow, so calculated, was found to be consistently and significantly higher than the sum of the calculated extraction steam flows to the eight heaters. That difference was considered to be the quantity of moisture discharged from the turbine into the extraction pipes and showed a high effectiveness of the moisture catchers. The amount of moisture thus indicated as removed in the wet extraction stages was about 10,000 to 12,000 lb of water per hr and was nearly constant irrespective of load. The amount of moisture removed from the 23rd-stage and the 24th-stage diaphragm was expected to be large but was not measured. An examination of the last-stage

---

[4] "The Design of Sewaren Generating Station and No. 1 Unit at Essex Station, Public Service Electric and Gas Company," by F. P. Fairchild, published in this issue of the Transactions, pages 247–265.

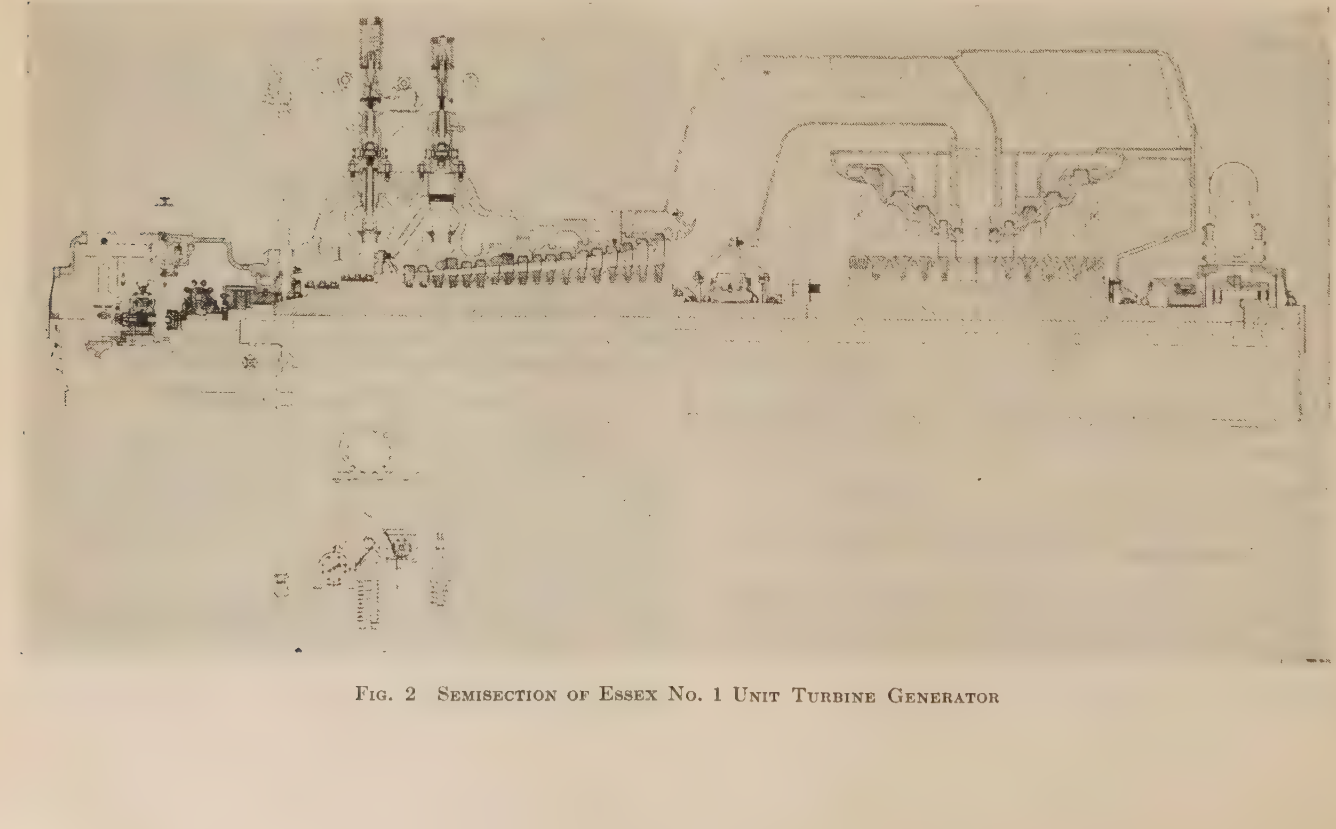


Fig. 2 Semisection of Essex No. 1 Unit Turbine Generator

buckets after 14 months of service confirmed this assumption since little erosion had occurred.

Heat-Rate Results

The heat rate of the turbine generator is defined as that quantity of heat supplied by the boiler (and the boiler feed and condensate pumps) necessary to produce 1 net kwhr of electricity. The net kilowatthour output is defined as the output at the terminals of the two generators minus the electrical inputs to the two generator fields.

Since the steam conditions and heater performance were not the same in every test, the test results have been corrected to a standard cycle to provide a uniform basis of comparison. The standard cycle differs from the test cycle principally in that:

1 The initial steam conditions were 1250 psig and 1000 F.

2 Exhaust pressures were 1.00 in., 1.50 in., 2.50 in., and 3.50 in. Hg abs.

3 Heater terminal differences and extraction-pipe pressure drops were assumed to be the same as in the turbine specification.

4 Attemperator flow was zero. (The attemperator was used only in 7 test points.)

The standard extraction cycle heat rates, based on test performance, are shown in Fig. 3 as a function of load for the four exhaust pressures. The points in Fig. 3 have been connected by valve loops, the shapes of which have been determined by previous tests on other turbines. The rise in heat rate at loads above 93 mw is caused by the opening of the internal stage valves which provide a by-pass for the steam flow around the 2nd and 3rd stages. The heat-rate curve at 1.50 in. Hg exhaust pressure is well located with eleven points, but it should be noted that the curves at 1.00 in., 2.50 in., and 3.50 in. Hg are drawn from only three test points with the shape determined by the 1.50-in. curve.

The standard extraction cycle throttle and condenser steam rates based on test performance are shown in Fig. 4. It may seem surprising that the throttle steam rate is about 4 per cent better at 35 mw than at 117 mw while the heat rate is about 9 per cent poorer at the lower loads. This apparent inconsistency is explained readily since the amount of steam extracted for feedwater heating changes from about 22 per cent to 32 per cent over the same range of load. The larger percentage of condenser steam flow at 35 mw has the effect of reducing the amount of throttle steam per kilowatthour at the expense of a less efficient cycle with its higher heat rate. This example shows that a low throttle steam rate is not necessarily compatible with a low heat rate. Had this turbine been operated nonextracting, the steam rate would have been the same for both throttle and condenser flows and would have a value between the two sets of curves shown in the figure. As a matter of interest, the feed-

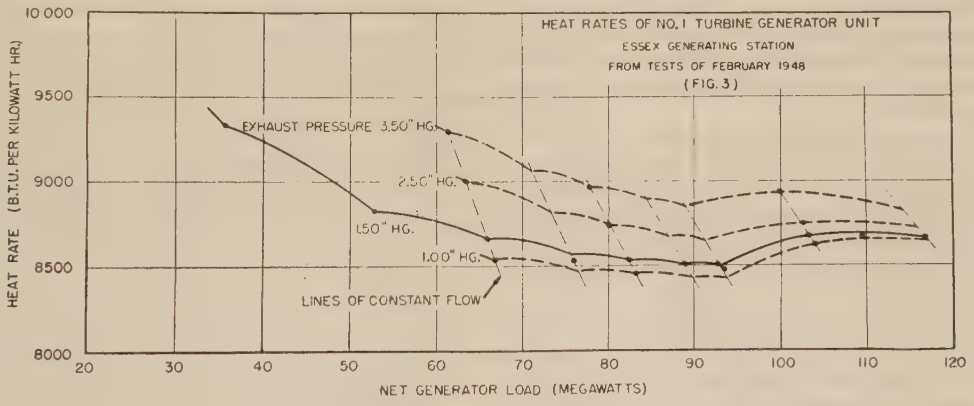


Fig. 3 Heat Rate of Essex No. 1 Unit Turbine Generator

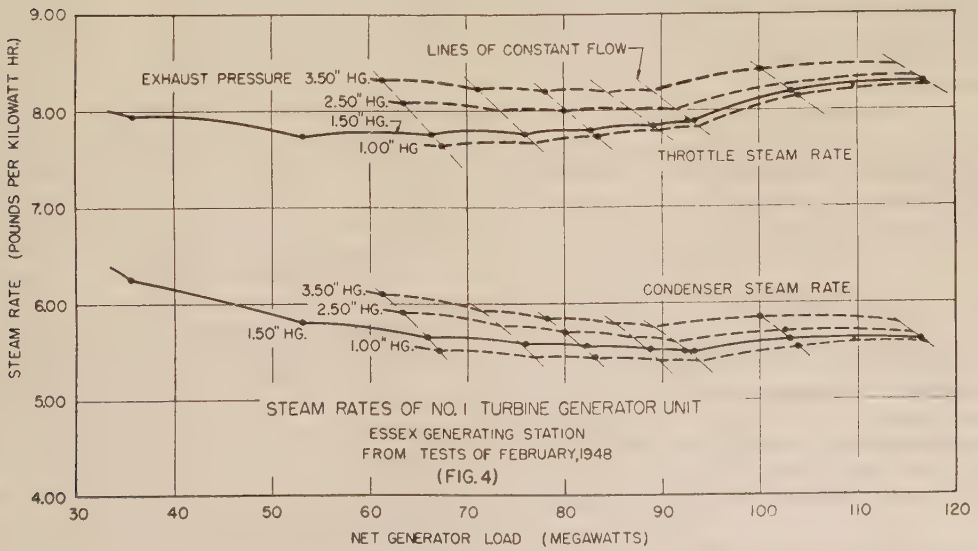


Fig. 4 Throttle and Condenser Steam Rates of Essex Unit No. 1 Turbine Generator

water temperature leaving the last heater was 477 F at the highest load.

ACCURACY OF THE TESTS

Extreme care was taken in the isolation of the unit, selection, installation, and calibration of the instruments, and constancy of test conditions. It was expected that the absolute accuracy would be about $^1/_2$ per cent, and the reproducibility somewhat better. The one check point at 93 mw differs from the first point by about 0.15 per cent. Since one point was made at the beginning and the other near the end of the test, there was no appreciable change of performance of the unit during test. There was no evidence of deposits in the steam path either by inspection or from unit performance.

GENERATOR TEST

The results of the generator preheating test will be reported elsewhere,[5] and will not be described here. The generator losses were of the magnitude predicted by the manufacturer and the maximum temperatures were well within the design limits.

CONCLUSIONS

At the best point, which was at 93 mw load, and at an exhaust pressure of $1^1/_2$ in. Hg abs, the heat rate of the turbine-generator and feedwater heaters was found to be 8498 Btu per kwhr for the standard cycle used to compare the test points. The corresponding thermal efficiency is 40.16 per cent. The standard cycle differed from the operating cycle by only a few tenths of a per cent.

At the rated exhaust pressure of $1^1/_2$ in. Hg abs, the heat rate averages 0.76 per cent better, and the best point is 1.4 per cent better than the guarantee.

Test data on the complete unit including boiler, turbine, feedwater heaters, and auxiliaries indicate an approximate over-all heat rate of 10,100 Btu per kwhr at the most economical point during winter conditions. The corresponding thermal efficiency is 33.79 per cent. With 13,500 Btu per lb coal, that heat rate corresponds to the consumption of 0.75 lb of coal to produce 1 net kwhr of electrical energy at generator terminals.

During the entire year of 1948, the station records show that the over-all unit heat rate was 10,500 Btu per kwhr at an average load of 97,800 kw for each hour of operation.

Tests of the operating characteristics have demonstrated that the unit operates satisfactorily with fast starts and high loading rates. Several load-dump tests followed by fast pickup of load have been made with excellent performance.[3]

The Essex No. 1 Unit has satisfied both the owner and the manufacturer with respect to efficiency, operating characteristics, and service record.

ACKNOWLEDGMENTS

The authors express their appreciation for the valuable assistance of Messrs. E. M. Kratz, P. C. Van Nordstrand, and M. W. Leonard in making the test and calculations.

Appendix

DESCRIPTION OF TEST INSTRUMENTS

The location of each of the test instruments is shown in Fig. 1.

All important temperatures were measured with chromel P-copnic thermocouples inserted in wells. A few temperatures were measured with copper-copnic thermocouples peened into pipe walls. All thermocouple potentials were measured with precision potentiometers. The condensate temperatures at the generator coolers were measured with precision-grade mercury-in-glass thermometers. All thermocouples and thermometers were calibrated before and after the test, except for the peened couples which necessarily were destroyed in disassembly. Temperatures were read at ten-minute intervals, with the exception of those at the throttle and at the discharge from the highest feedwater heater, which were read every 5 min.

Pressures above 25 psia were measured with dead-weight gages, those between 5 psia and 25 psia with mercury-filled manometers open to atmosphere, and the condenser pressures were measured with mercury-filled absolute gages. All gages, weights, and manometers were checked carefully before the test. For convenience and accuracy, all turbine pressures at levels below 25 psia were measured with air-filled connecting pipes. This was accomplished by bleeding a very small amount of air into each pipe.

The condensate flow was measured with either a 4.335-in. or a 2.794-in-diam flow nozzle. Both were calibrated before the test by Prof. S. R. Beitler. The pressure difference was indicated on a mercury-filled manometer read at one-minute intervals.

As an independent check on the throttle flow, the pressure difference across the uncalibrated station steam-flow nozzle was read at one-minute intervals. However, the water legs could not be determined accurately, and the measurements were not entirely consistent or reliable. This trouble was corrected after the test.

The condensate flows to the gland-seal tank and from the high-pressure-turbine packings were measured with displacement-type integrating water meters. The flow quantity was recorded every 15 minutes.

The flow to the attemperator was measured with the station nozzle. Since the flows were small, a dibromoethane (specific gravity = 2.17) filled manometer was used to indicate the pressure difference. Readings were made at five-minute intervals. The maximum attemperator flow was $3^1/_2$ per cent of the throttle flow.

Slag-tap and boiler-access door cooling flow was measured with a flow nozzle and a mercury-filled manometer. Readings were made at five-minute intervals.

The flow of condensate through the generator coolers was measured by thin-plate orifices. The coefficients were taken from the ASME fluid meters report.[6] The pressure differences were indicated on mercury-filled manometers and read at five-minute intervals. In order to obtain a larger temperature difference across the main generator coolers at several test points, the arrangement of the station piping made it necessary to by-pass some condensate around the auxiliary-generator coolers, which prevented measurement of the cooler flow. For that reason, only a few loss measurements could be made on the auxiliary generator.

The leakage flow from the second section of the high-pressure packing was measured with a thin-plate orifice and a mercury-filled manometer, read at five-minute intervals. The leakage flow from the third section of the high-pressure packing was measured with an impact tube placed at the axis of the pipe and a pipe-wall static pressure tap. The pressure difference was indicated on a draft gage and read at five-minute intervals. This method of measuring the flow was necessary, since there was not sufficient pressure available for an orifice measurement.

Storages in the hot well and heaters were measured by observing calibrated sight glasses at 15-minute intervals.

The electrical output of each generator was measured with three single-phase indicating wattmeters read at one-minute intervals. The voltage and current in each phase were observed at two-minute intervals. The power input to the two fields was measured with voltmeters across the slip rings and milliammeters across shunts.

---

[5] The generator preheating test is expected to be the subject of an AIEE paper by F. W. Gay and H. D. Taylor.

[6] "Fluid Meters, Their Theory and Application," THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, New York, N. Y., fourth edition, 1937.

All ammeters, voltmeters, and wattmeters were calibrated before and after the test, and the current and potential transformers and the shunts were calibrated before the test. Check readings were made on the station integrating watthour meter which was especially equipped for the test with a high-speed register.

The tests followed the procedures outlined in the ASME Power Test Code. The Keenan and Keyes Steam Tables[7] were used to determine steam properties.

Before each test, the turbine and heater piping was carefully checked to make certain that the cycle used was the one shown in Fig. 1, and to check the isolation of the unit. Isolation was made positive in every connecting pipe not necessary to the cycle by inserting blind gaskets between flanges or by closing two valves in series and opening a "telltale" drain valve between them to prove that no flow passed through the pipe. A single valve was used for only one connecting line which went to the condensate head or surge tank. That valve was checked carefully for tight closing before, during, and after the test. It was not possible to double-valve this line from the standpoint of safety.

The losses of each generator were obtained by measuring the temperature rise and quantity of the condensate used to cool the hydrogen in the main generator, and the air in the auxiliary generator. Since the four coolers in each generator are connected in parallel and it was practical to make only one flow measurement for each generator, it was necessary to adjust the flow through each cooler to give the same temperature rise. The hydrogen pressure was maintained at 15 psig, except for the two highest load points where it was held at 25 psig.

All test points except one ($8^1/_2$ admission point) were made at a valve-intercept point. That point occurs just prior to the opening of the next higher valve. The intercept point could be set accurately by opening the governor until all lost motion is taken up in the cam-and-roller assembly of the next higher valve. Since a force of only a few hundred pounds was necessary to take up this lost motion and several thousand pounds to lift the valve away from the seat, a check of the force on the roller with a pinch bar permitted accurate setting of the valve within 2 or 3 thousandths of an inch. This method is considered to be the most accurate since a measurement of valve lift will vary slightly with different loads on the turbine as the temperatures of the valve mechanism and steam chest change. The valve-roller tightness and the position of governing mechanism were checked several times during each test to make certain the valves had not changed position. The governor was very stable and it was not necessary to reset the valve position during any test. During tests, the governor was set on the "load limit" which essentially kept the governor from increasing the load but still left control for reducing the load by normal governor action if an emergency should arise.

For different tests, the exhaust pressure at the condenser neck was held at $3^1/_2$ in., $2^1/_2$ in., $1^1/_2$ in. Hg abs, and the lowest that could be obtained. For the higher values, the pressure was held by bleeding air from the atmosphere into the condenser. This method is simple and permits rapid and close control of the exhaust pressure. It was usually held within 0.01 in. Hg.

Tests were made with an exhaust pressure of $1^1/_2$ in. Hg abs at each of the nine admission points, at the $8^1/_2$ admission point, and a check test was made at the 7th admission. Other tests were made at exhaust pressures of $3^1/_2$ in., $2^1/_2$ in., and best (about 1.1 in.) at the 3rd, 5th, and 8th admission points.

The initial steam conditions were held as closely as possible to the rating of 1250 psig and 1000 F. At the lighter loads, the boiler characteristics resulted in lower throttle temperatures. The throttle temperature usually was held constant within ±10 F, and the throttle pressure within ±10 psi.

All test points at exhaust pressures of $1^1/_2$ in. Hg abs were of 2-hr duration. Those at other exhaust pressures were 1-hr tests. To insure constant conditions, the load was set and held for 1 hr before the test period.

Discussion

Adolf Egli.[8] The 100,000-kw turbogenerator tested is a brilliant example of progress in modern power-generating equipment. As regards steam-power-cycle development, this unit represents the highest perfection to date of the regenerative-feed-heating cycle. The writer is struck by the elegant simplicity of the layout of the turbine flow path as well as with the completeness of the regeneration system applied.

The thermodynamic perfection attained is astounding; 40.16 per cent thermodynamic efficiency of the turbogenerator unit and 33.79 per cent thermal efficiency of the complete power unit, including boiler and auxiliaries, are indeed excellent values for a plant without the complication of a reheat cycle.

Remarkable is the combination of a flat heat-rate curve with great overload capacity which has been attained by a clever arrangement of throttle valves and by-pass valves regulating the throttle steam flow. Between maximum load and half maximum load, the heat rate (at $1^1/_2$ in. Hg back pressure) varies only 1.7 per cent, the best value being obtained midway between these points at the most important output. Even at 30 per cent maximum capacity, the heat rate is only 9.5 per cent above the best value.

Of the test instrumentation, the dead-weight gages used to measure the high steam pressures, are most intriguing to the writer.

Would the authors be in a position to give a detailed description of the design and operation of these gages? Precise measurement of the high steam pressures on a turbine in operation is a difficult problem, and the proper application of the dead-weight system would appear to be a valuable step in improving test precision.

In particular, the writer is also interested in the accuracy attained with condensate-flow-nozzle measurement.

A detailed heat balance of the unit tested would also be most informative. By publishing this information the manufacturer and the Public Service Electric and Gas Company would indeed render valuable service in the general advancement of the art of power generation.

Authors' Closure

The authors thank Professor Egli for his comments on the design and application of the Essex turbine generator.

It has been the practice of the authors' company to use dead-weight gages in the tests of steam turbines since they provide the most practical fundamental and accurate type of pressure measurement. It is difficult to use this type of gage when the pressure fluctuates rapidly. However, modern methods of pressure control in generating stations are sufficiently good to permit the use of deadweight gages. Although more work is involved in its installation and use, the increase in accuracy is believed to justify the additional time and expense. The area of our dead-weight-gage cylinder is 0.1250 sq in. (a diameter of about 0.4 in.) and is larger than the usual dead-weight gage or dead-weight-gage tester. The sensitivity of about $^1/_2$ in. of water and the practical advantage of free plunger travel is believed to overweigh the inconvenience of the correspondingly large weights.

---

[7] "Thermodynamic Properties of Steam," by J. H. Keenan and F. G. Keyes, first edition, twelfth printing, John Wiley & Sons, Inc., New York, N. Y., 1946.

[8] Professor of Mechanical Engineering, Towne Scientific School, University of Pennsylvania, Philadelphia, Pa. Mem. ASME.

The gages are oil-filled and, in steam-turbine tests, are usually operated with water-filled lines, necessitating the usual water-leg correction. In operation, the pan is loaded with weights until the pressure is balanced, the weight pan being rotated by hand to remove any friction effect. Arrangements have been used with the pan loaded with a spring or balance arrangement of about 10 psi capacity to avoid the use of small weights. Such devices have also been used for the control of pressure with the rotation of the plunger accomplished by an air turbine. The radial clearance between the cylinder and plunger is held to about 0.0001 in. and the parts are usually made of nitraloy with lapped surfaces.

There is a question as to whether the area of the cylinder or of the plunger or some intermediate value should be used. With these close fits, the error involved when an average area is used cannot be more than about 0.05 per cent. Calibrations by the Bureau of Standards and others confirm this opinion.

The two nozzles used in this test were made by the Bailey Meter Company and installed with straightening vanes in a pipe of suitable length. The two assemblies were calibrated by Prof. S. R. Beitler and are believed to represent the true coefficient within ± 0.2 per cent. Each nozzle was used only in the Reynolds number range of constant value of the flow coefficient. The smaller nozzle was used in the Reynolds number range of 1,200,000 to 1,600,000, and the larger from 1,200,000 to 2,200,000. While these ranges require extrapolation of the calibration to somewhat higher values than could be reached in the calibration, that extrapolation was in the region of the constant value of the flow coefficient and is well justified by experience.

The addition of heat balances at three loads was considered when this paper was prepared, but it was decided to omit these items because of lack of space.

Frictional Characteristics of O-Rings With a Typical Hydraulic Fluid

By L. E. CHEYNEY,[1] W. J. MUELLER,[2] and R. E. DUVAL,[2] COLUMBUS, OHIO

O-ring packings employed in aircraft hydraulic systems are subject to numerous operating variables, which affect the friction involved in their operation. A method for studying the friction is described, and the effects of several variables—pressure, time delay, squeeze, stroke speed, ring size, and surface finish of the moving metal part—have been determined.

Introduction

During the past several years there has been a strong trend in the aircraft industry toward the use of hydraulic systems, as a result of increases in the size of planes, where the required forces prohibit manual operation of many controls. The hydraulic system has an advantage over other possible systems because it assists in the movement of controls, but it still maintains the "feel" which is considered highly desirable.

Among the small but important parts of a hydraulic system are the packings. Earlier systems in use for aircraft employed one or more of the well-known V-ring, U-cup, or compression-type packings. The O-ring is a fairly recent development in the packing field, which has been shown to have many advantages over the older types.

To prevent sticking of moving parts and excessive wear on packings, the friction between the moving parts and the packing should be held to a minimum. It would be expected that the early compression-type packings would cause high friction because of their design. O-rings offer an opportunity to reduce this friction because of a much smaller area of contact with the moving part.

Previous published work on the subject of friction is meager. Several recent articles (1, 2, 3, 4, 5, 6, 7, 8)[3] give limited data concerning the effect on packing friction of various characteristics of the hydraulic system, while the general frictional characteristics of rubber are treated in some previous publications (9, 10, 11). While these articles cover a part of the field, some of them freely admit that there is much work yet to be done.

Experimental Procedure

An apparatus was designed so that friction caused by an O-ring could be measured under controlled conditions, with one factor varied at a time. In the apparatus employed, an actuating rod was drawn through a stationary O-ring. Frictional forces were transmitted from the ring, through a bellows arrangement, to a recording engine indicator.

The entire apparatus could be considered as composed of several individual units as follows:

1 The nucleus of the machine was a hydraulic press, to the ram of which was attached an actuating rod (SAE 4340 steel, Rockwell hardness 40 C). Ram speed of the machine could be varied from 85 to 300 ipm on the upstroke, and 60 to 215 ipm on the downstroke. A timing device was installed to allow an automatic variable time delay between strokes.

2 The hydraulic pressure unit consisted of a pump and reservoir with accessory equipment such as time switches, tubing, etc. The variable-flow pump employed was capable of producing 3000 psi fluid pressure. The oil reservoir, a large box made from steel plate, held about 50 gal of oil and was provided with cooling coils.

3 The assembly for holding the O-ring under fluid pressure consisted of a section of stainless-steel tubing with a cap screwed on each end. The end of each cap contained a hole, through which the actuating rod passed. One end of the tube was sealed by the O-ring, contained in a machined groove inside the cap. The other cap contained no packing but had a lap fit around the rod with a clearance of about 0.001 in. (The friction from this lap fit was found to be negligible.) The hydraulic pressure was applied to the inside of the sealed tube.

4 The friction-measuring unit consisted of a bellows connected to an engine indicator by means of copper tubing, the entire unit being filled with oil and sealed. The frictional force was transmitted from the ring to the bellows by a special cap, which in turn transmitted it through the oil to the recording engine indicator, which recorded the friction continuously.

In any work of this type the accuracy of test results should be considered. It is felt that the measuring unit is quite accurate because of its basic design. The spring on the engine indicator can be calibrated with dead weights and this is done at frequent intervals. A small error is involved in reading the engine indicator cards because a small distance between lines on the card corresponds to several pounds friction. This error is of the order of 3 or 4 lb maximum; it could probably be reduced if necessary by constructing a special micrometer-type scale for reading the cards. Assuming the measuring unit to be quite accurate, the accuracy of the work would then be limited only by the degree to which the friction would reproduce itself.

An effort was made to evaluate what were considered the more important variables, as indicated by previous work (1, 2, 3, 4). This study was concentrated upon one hydraulic fluid (Army-Navy Specification AN-VVO-366B). All of the work was performed using one typical ring size (-28 as described in Army-Navy Specification AN6227), except the study of ring-size effect. Testing was done at room temperature. All of the rings used in this study were aged in hydraulic fluid before testing, according to Army-Navy Specification AN-P-79.

Experimental Results

Previously published data have recommended that the O-ring groove depth be 10 per cent less than the free diameter of the ring (6). The purpose of the shallow groove is to make certain that the ring is subjected to a positive squeeze at all times. This positive squeeze is necessary if the ring is to seal at low pressures where the pressure is not great enough to deform the ring and force it against the diametral clearance between the moving

---

[1] Formerly Assistant Supervisor, Resin and Rubber Research, Battelle Memorial Institute.

[2] Research Engineer, Battelle Memorial Institute.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Rubber and Plastics Division and presented at the Annual Meeting, New York, N. Y., November 28–December 3, 1948, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-64.

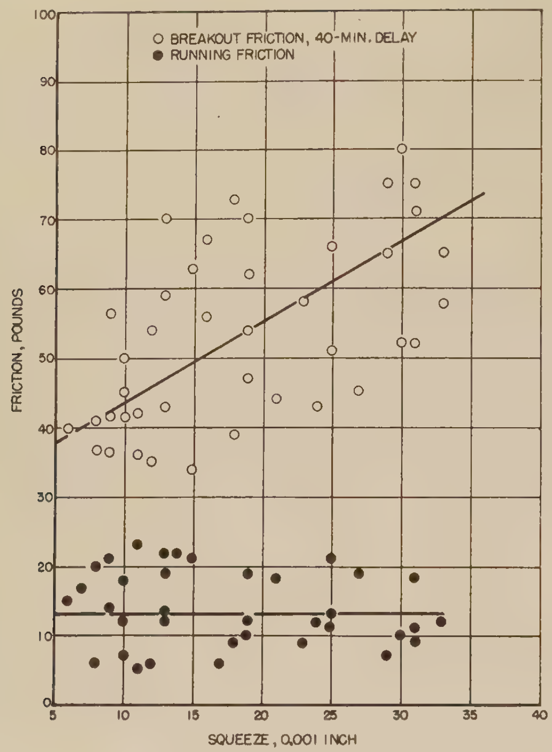


Fig. 1 Effect of Squeeze on Friction

(2.8–3.2 microinches surface finish, 300 ipm stroke speed, 500 psi pressure.)

and stationary parts. Army-Navy Specification AN-P-74 designates the minimum squeeze to be applied to an O-ring; this squeeze is approximately 10 per cent of the cross-section diameter of the ring.

Breakout- and running-friction determinations were made under various conditions of pressure and time delay, in addition to variation of squeeze. The result of a large number of tests shows that the running friction is largely independent of squeeze over the range covered. There is some tendency at higher pressures for the running friction to increase slightly with squeeze. A different situation exists in the case of breakout friction, where an increase in squeeze results in an increase in friction. Fig. 1 shows typical data of friction versus squeeze for running and breakout friction. The points shown are chosen as representative of about 2500 determinations. The range of results, as shown by the figure, is approximately ±15 lb for the breakout friction and ±8 lb for the running friction.

The effect of hydraulic pressure has been noted previously (1, 2). As there is a strong trend toward higher pressures in the newer aircraft systems, this is a very important factor. The effects of 500, 1000, and 1500 psi pressure were checked in this study. It was found that an increase in pressure resulted in a small increase in running friction and a large increase in breakout friction. Figs. 2 and 3 illustrate this point. The data shown here are typical of about 700 determinations. It can be seen that the accuracy is approximately ±15 lb for the breakout friction and ±10 lb for the running friction.

It has been generally felt that the breakout friction would be higher as time delay between strokes increased. Experimental results confirmed the original belief for time delays of less than 40 min. Time delays of more than 40 min, however, did not increase the friction appreciably.

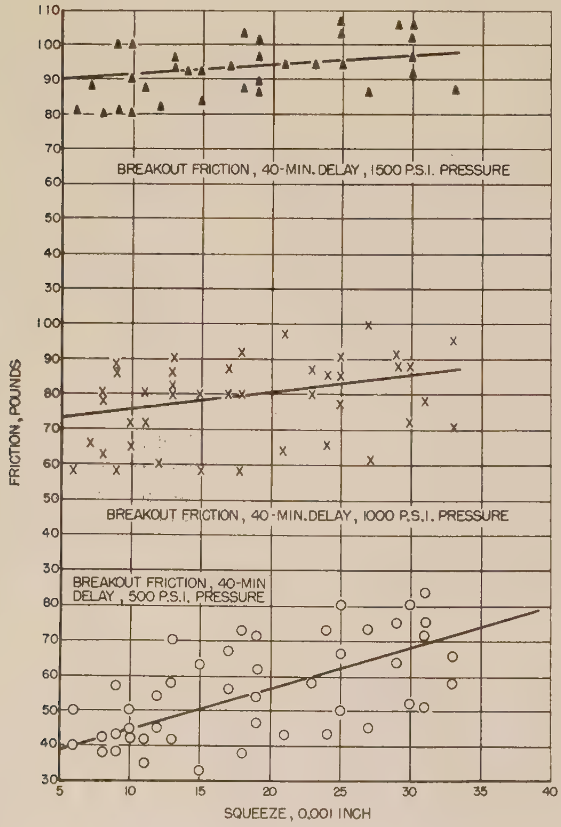


Fig. 2 Effect of Pressure on Friction

(2.8–3.2 microinches surface finish, 300 ipm stroke speed.)

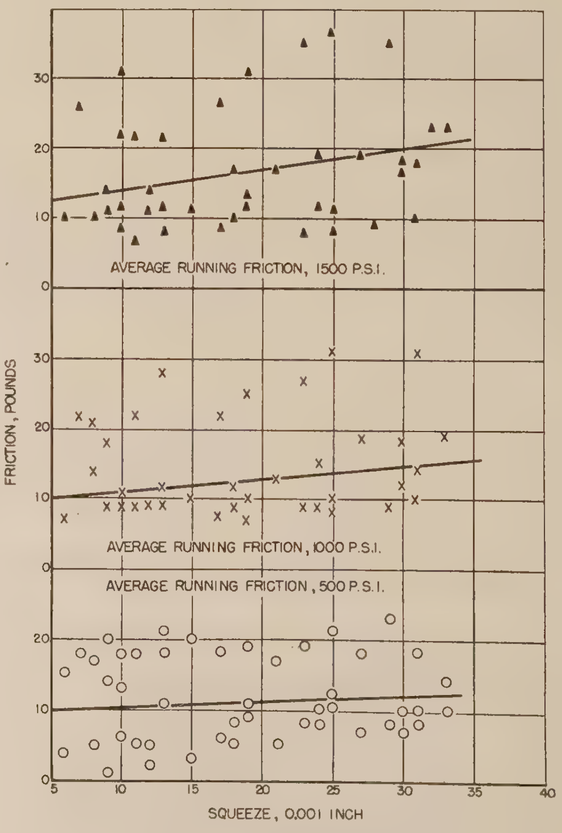


Fig. 3 Effect of Pressure on Friction

(2.8–3.2 microinches surface finish, 300 ipm stroke speed.)

Running friction was not expected to change with time delay between strokes. In this case also, the experimental data confirmed the original belief. Fig. 4 shows data, typical of about 1600 determinations, on the effect of time delay on the two types of friction. The accuracy of reproduction is about ±10 lb.

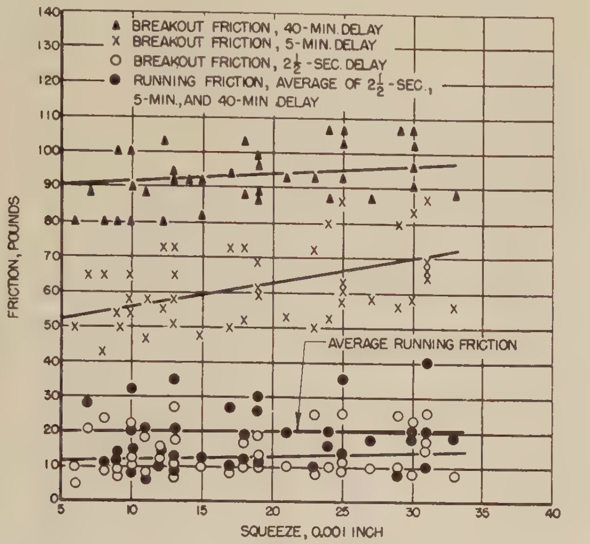


Fig. 4 Effect of Time Delay on Friction
(2.8–3.2 microinches surface finish, 300 ipm stroke speed, 1500 psi pressure.)

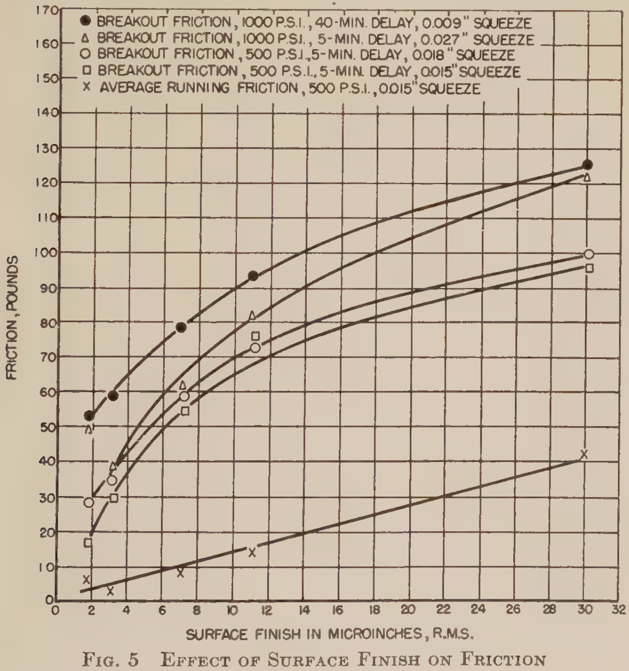


Fig. 5 Effect of Surface Finish on Friction
(300 ipm stroke speed.)

The importance of roughness of the surface which rubs against the ring has been recognized by other investigators (1, 2). Army-Navy Specification AN-P-74 requires a surface finish (roughness) of 5–15 microinches root mean square (rms). (This is to be changed by eliminating the minimum requirement and retaining the 15-microinches rms maximum roughness.) In order to investigate this factor more thoroughly, a series of 4 rods was constructed. These rods were ground to finishes of 2.8–3.2, 6–8, 10–12, and 16–18 microinches rms. All four rods were evaluated with the same ring to eliminate any variability from that source. These four rods represent a range of surface finishes not only covering, but also above and below the present specification limits. Fig. 5 shows typical curves for this series. It can be seen that there is a definite increase in friction with an increase in surface roughness.

An additional rod with a surface finish of 1.7–2.0 microinches rms was constructed and evaluated later in a manner similar to the four rods previously studied, although the same ring was not used. The data obtained for this rod are also included in Fig. 5 and show good agreement with the other values. The accuracy of these data is considered to be approximately ±10 lb.

It has been reported that a chemical surface treatment was being used on some hydraulic parts. Several advantages were claimed for this treatment, particularly reduction of friction, because of better retention of an oil film on the surface. In order to investigate this point, a special rod was constructed and treated by the commercial process. This rod had a surface finish of 10 microinches rms before treatment and 30 microinches rms after treatment. This rod was included in the series showing the effect of surface finish on friction, Fig. 5. The frictional results obtained were apparently directly proportioned to the roughness of the finished rod.

A limited amount of work was done to determine the effect of stroke speed. Figs. 6 and 7 show these data. The running friction was almost independent of stroke speed over the range tested. An increase in breakout friction was noted with an increase in stroke speed at 1000 psi, but at other pressures the friction was independent of stroke speed. There are indications that factors other than stroke speed, for example, pressure, are more significant in determining the friction in a system.

The effect of ring size on friction has been reported by Pearl (2). This author showed the friction to be a function of pressure and projected area of ring exposed to pressure. The area of contact of a packing and moving surface would be expected to affect the amount of friction.

Four sizes of O-rings were given an evaluation to determine the effect of ring size. The ring sizes were —19, —23, —28, and —29 as described in Army-Navy Specification AN6227. The actual sizes are given in Table 1.

TABLE 1 SIZES OF O-RINGS

Ring	Cross-sectional diam, in.	ID, in.	OD, in.
—19	0.139	0.984	1.262
—23	0.139	1.234	1.512
—28	0.210	1.475	1.895
—29	0.210	1.600	2.020

The sizes were chosen to show the effect of diameter at constant cross section, using two different cross sections. Graphs of friction versus squeeze, at different conditions of pressure and time delay, where the data from all four rings were plotted on one graph, show that under some conditions one ring size gave the greatest friction while under other conditions another ring caused the greatest friction. Although the friction of all four rings followed the same general pattern, it could not be concluded that any one ring size consistently gave higher friction than another ring size, at least in the range of dimensions investigated.

Reproducibility of results has been checked in two different ways. One ring from each of several manufacturers was evaluated thoroughly and the comparative results recorded, then five rings from one manufacturer were evaluated in succession. Fig. 8 compares the reproducibility between rings from one manufacturer with that of rings from different manufacturers. It can be seen that the distribution of points in the two curves is very nearly the same, that is, reproducibility between rings of different manufacturers is approximately the same as reproducibility be-

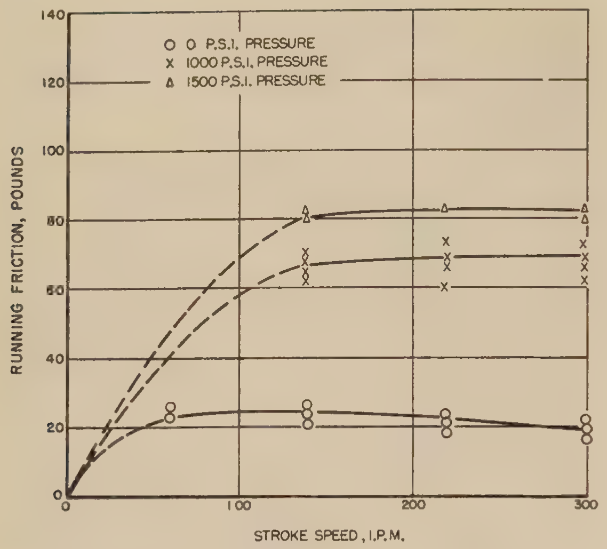


FIG. 6 EFFECT OF STROKE SPEED ON FRICTION
(2.8–3.2 microinches surface finish, 0.017 in. squeeze.)

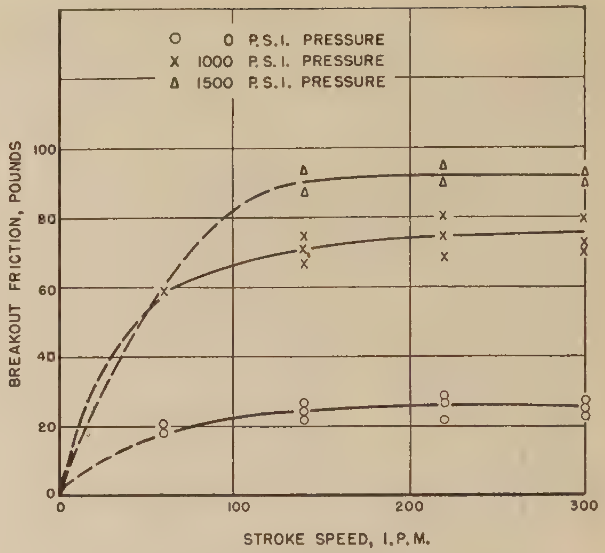


FIG. 7 EFFECT OF STROKE SPEED ON FRICTION
(2.8–3.2 microinches surface finish, 0.017 in. squeeze.)

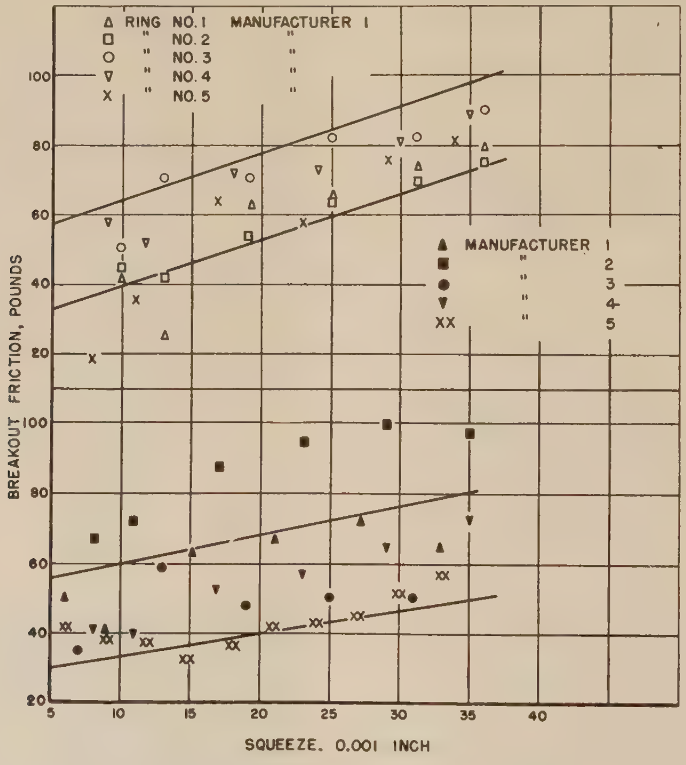


FIG. 8 REPRODUCIBILITY OF FRICTION OF O-RINGS
(2.8–3.2 microinches surface finish, 300 ipm stroke speed, 40-min delay, 500 psi pressure.)

tween several rings from one manufacturer. It can also be seen that the friction curves for the different manufacturers' rings are approximately the same within the accuracy of our testing. From this it can be concluded that all of the manufacturers whose aircraft hydraulic packings have been checked produce rings of similar frictional characteristics, although there are insufficient data to justify a general conclusion that all are the same.

SUMMARY

Certain design and operating variables in aircraft hydraulic systems have been studied in relation to friction.

It is possible to summarize the resulting data in a fairly comprehensive form. Table 2 summarizes the effect of the variables studied on breakout and running friction.

ACKNOWLEDGMENT

The work described in this paper was carried out in connection with a research project sponsored by the Bureau of Aeronautics, Navy Department. Any opinions expressed here are those of the authors and do not necessarily represent those of the Navy Department. Acknowledgment is also made of the assistance rendered by other Battelle personnel.

TABLE 2 VARIATION OF BREAKOUT AND RUNNING FRICTION WITH CHANGES IN HYDRAULIC SYSTEM

	Effect	
Design or operating change	Breakout friction	Running friction
Increase squeeze	Increase	No effect at low pressure (500 psi). Slight increase at high pressure (1000, 1500 psi).
Increase time delay between strokes	Increase	No effect
Increase hydraulic pressure	Increase	Increase (slight)
Increase roughness of moving rod	Increase	Increase

BIBLIOGRAPHY

1 "Hydraulic O-Ring Packing for Sealing Cylinders and Pistons," by R. E. Allen, *Product Engineering*, vol. 18, 1947, pp. 163–166.

2 "O-Ring Seals in the Design of Hydraulic Mechanisms," by D. R. Pearl, SAE Quarterly Transactions, vol. 1, 1947, pp. 602–611.

3 "Aircraft Hydraulic Systems—Factors Influencing Their Performance," by L. E. Cheyney and T. J. McCuistion, *Mechanical Engineering*, vol. 70, 1948, p. 675.

4 "Research on Aircraft Hydraulic Systems," by T. J. McCuistion, F. E. Clark, R. A. Clark, and L. E. Cheyney, SAE Quarterly Transactions, vol. 2, 1948, p. 227.

5 "Investigation of 10° Undercut Groove for O-Ring Packing," by B. R. Vennum, Douglas Aircraft Company Report SM-20063, Dec. 10, 1945.

6 "Performance of Linear LT2-70 Hydraulic O-Ring Packing Used With Chrome Retanned Leather Back-Up Rings at 3000 Psi Pressure," by E. R. Broadbent, Douglas Aircraft Company Report SM-11809, April 4, 1946.

7 "Performance of Houghton No. 2216 Back-Up Rings With O-Rings on Large Struts at 3000 Psi Pressure," by E. R. Broadbent, Douglas Aircraft Company Report SM-11155, Jan. 1, 1946.

8 Letter from F. G. Newlon to L. E. Cheyney, Army Air Forces, Washington, D. C., Aug. 15, 1947.

9 "The Coefficient of Friction of Rubber," by J. B. Darieux, *Rubber Chemistry and Technology*, vol. 8, 1935, pp. 441–442.

10 "Frictional Properties of Rubber," by F. L. Roth, R. L. Driscoll, and W. L. Holt, National Bureau of Standards *Journal, of Research;* vol. 28, 1942, pp. 439–462; *Rubber Chemistry and Technology*, vol. 16, 1943, pp. 155–177.

11 "The Coefficients of Friction Between Rubber and Various Materials, Part II, Gripping Friction of Rubber Belting," by R. Ariano, *India Rubber Journal*, vol. 79, 1930, pp. 56–58; *Rubber Chemistry and Technology*, vol. 3, 1930, pp. 286–292.

Discussion

G. E. Davisson.[4] The authors are to be congratulated on the excellent work that is summarized in the paper. A short history of this testing program will reveal its vital importance.

History of Testing Program. During the last war the Army Air Force and Navy Bureau of Aeronautics added the winterization requirements, calling for operation of aircraft hydraulic equipment at temperatures from —65 to +160 F, to the AN Specifications covering this equipment. This requirement made obsolete all of the old components of hydraulic equipment that were designed for operation to a minimum temperature of —40 F. It also made necessary an intensive program of development and testing by the services and the aircraft industry. The use of the original winterized O-rings with AN-VV-O-366b winterized hydraulic fluid resulted in an epidemic of so-called spiral failures and other miscellaneous types of O-ring failures.

In 1944 the Technical Advisory Committee on Hydraulic System Materials was formed. This committee is composed of representatives of the two services, and the manufacturers of aircraft, of hydraulic fluid, of packings, of hose, and of miscellaneous hydraulic equipment. The function of this committee is to assist the services in their program to improve performance of aircraft hydraulic systems. The committee makesre commendations for and co-ordinates the results of tests made on O-rings, hydraulic fluid, and all types of hydraulic equipment by industry.

As the result of a great deal of test and development work by the services and by manufacturers of aircraft hydraulic fluid and packings, with help and advice of the Technical Advisory Committee on Hydraulic System Materials, there had been by 1946, a gradual improvement in the quality of hydraulic fluid and packings and in the compatability of one with the other. However, there was urgent need for a comprehensive study of the fundamentals of packing and fluid behavior, which neither the services nor industry was in a position to undertake. Battelle Memorial Institute was found to be the organization that could undertake such a study. Battelle initiated test programs for both the Air Force and the Navy, and personnel of Battelle, active in this work, were made honorary members of the Technical Advisory Committee.

The present paper is based upon data obtained in this testing program. This information is valuable because it helps to build a foundation of scientific facts which will permit the services and industry eventually to eliminate the causes of packing failures in aircraft hydraulic systems. A great deal of progress has been made on this program, but a lot remains to be done. However, continued co-operation between the organizations involved eventually will insure that our pilots will fly better safer aircraft.

O-Ring Friction in Industrial Applications. The Battelle tests were for typical aircraft installations. For industrial applications, automotive type of cylinder oil is used more frequently than AN-0-366 hydraulic fluid. This is due to its ready availability, especially for refilling the systems. There will be less friction when cylinder oil is used. In most cases an O-ring made from a special industrial compound will be required for use with automotive cylinder oils. Use of brass or aluminum for piston rods or cylinders in place of the SAE 4340 steel will increase the friction, all other variables being the same.

O-Ring Friction in Pneumatic Systems. The friction in well-lubricated aircraft and industrial pneumatic installations normally will run higher than in comparable hydraulic systems. Friction is extremely high on poorly lubricated pneumatic installations where O-ring packings are used. Normally, O-rings are not recommended in unlubricated pneumatic installations, due to the excessive friction and the resultant high percentage of ring failures.

B. P. Graves.[5] O-rings properly applied, are quite effective in preventing leakage between sliding surfaces, at least for a time. The important question to consider is—"How long will they last?". The writer has been seeking for some time to find a reliable answer to this question within our own personal applications and research, or in any of the papers on the subject of O-rings which have come to hand.

In discussing this subject with many hydraulic engineers and users of hydraulic equipment, the life of sliding O-rings has been questioned seriously, and the consensus of opinion seems to be that their life is too short. So instead of presenting a discussion on this paper, the writer finds it almost necessary to question the authors as follows:

Can the life of O-rings be extended to a practicable degree by practical methods, or ordinary shop methods? Of particular interest to us would be the effect on life of O-rings, of such matters as relative smoothness of the sliding surfaces, the condition of the grooves into which these O-rings fit, the composition of the O-ring and its hardness, amount of initial compression, working pressures, and speed of oscillation.

The several questions sum up to this one particular issue—is the life of an O-ring to be measured by the number of starts, the total distance traveled, or is it mainly a question of time under pressure?

---

[4] Walker Manufacturing Company, Racine, Wis.

[5] Director of Design, Brown & Sharpe Manufacturing Company, Providence, R. I. Fellow ASME.

We are particularly interested in the O-ring application because it offers a tantalizing picture of extreme compactness, high initial efficiency, and economy.

Answers to the foregoing questions would be of great value to the application engineer and to the user.

L. S. Linderoth, Jr.[6] It is most interesting to note the indications of uniform performance of rings made by different manufacturers. The data in the curves of running and breakout friction should be very helpful to the design engineer.

If future work is contemplated to carry on and amplify these data the writer would like to suggest a series of experiments to determine the possible variation in friction with different rod materials such as chrome plate, stainless steel, nonferrous alloys, and so on. From the designer's point of view it would be helpful to have the friction expressed in terms of rod or cylinder circumference. It would also be helpful to know whether the present AN specifications on the O-ring material are optimum for all types of service. Personally, the writer believes that for industrial use a better material can be compounded, since for one thing the extreme temperature requirements of the AN specifications are seldom encountered.

He has often wondered whether a rod or cylinder surface could be made too smooth, and perhaps the results indicated in Fig. 5 of the paper, on the running friction of the 1.7–2.0 rms rod bears this out. Notice that the point indicates a higher friction than the extended curve.

Personal experience has indicated that long time delays, several days to several months in duration, give extreme breakout friction values. This paper does not indicate whether any experiments considerably over 40 min duration were made. This is an important factor as relating to storage life of completed units and deserves considerable attention.

Were the points on the various curves measured with the rod going in one direction only or were directions measured and recorded? If both directions were recorded was there a difference between frictions with a dry and an oily rod?

On the measurement of the breakout friction with variation in rod speed, does not acceleration play a larger part than change in friction? With the bellows, fluid, and engine indicator parts all having appreciable mass, some question may arise as to the accuracy of the measurement of nonstatic loads. It may be desirable in future experiments to use a measuring means independent of mass in the measuring instrumentalities.

All tests in this paper were carried on at room temperature, presumably 70 F. It has been the writer's experience that most operating equilibrium temperatures are on the order of 150 F, unless special cooling is introduced. It would be helpful to know what effect temperature would have on the running and breakout friction with the time delay starting at the elevated temperature.

B. R. Teree.[7] The study of all packings used in aircraft hydraulic systems regarding their frictional behavior due to change in squeeze, time delay between strokes, pressure and roughness of moving surfaces has always been of the utmost need. For years the aircraft hydraulic designers and engineers have been searching for such useful data to improve the operation of their equipment by reducing over-all friction. The discussion covered in this paper gives a clear-cut analysis of the frictional characteristics of O-rings, most commonly used in the aircraft industry at this time, and the authors along with the Bureau of Aeronautics, should be complimented for this presentation.

In almost all aircraft hydraulic-equipment design work, there are two major points or locations where friction data on packing installation is badly needed. These two points are rod-gland-seal installation; and piston-head-seal installation. It is also pointed out that most of the serious problems in industry are with seals which are installed at rod ends of cylinders, where the leakage can be detected very easily.

The authors have discussed the investigation of this gland seal very thoroughly. In the test setup employed by the authors the O-ring does not change position at the start of the stroke. In an actual aircraft piston-ring seal the O-ring is at the side of the groove next to the pressure prior to starting the stroke. Therefore the O-ring must move to the opposite side of the groove as the pressure is applied, and as the stroke starts. This test information and data may not correlate with actual service experiences on piston-head seals. It is believed that in actual operating conditions the breakout friction will be less due to the rolling of the O-ring before the actual starting of the stroke.

It is surprising that in no place in this paper is there mention of friction tests at zero or very low pressures. Installations with such requirements are common in the aircraft industry.

There is also no mention of friction of O-rings with the use of leather back-up rings. This is understandable because there is an increased amount of work and testing involved for such an investigation.

It is stated in the paper that in the range of dimensions investigated, there was no appreciable difference in friction due to O-ring sizes. It is hoped that this will not be misinterpreted or confusing to designers because there must be a difference in friction due to change in size of O-ring as friction should be directly proportional to contact area.

The information covered in this paper is very valuable to the aircraft industry and also to all industrial hydraulic applications. It is important that it is leading to the discovery of causes of failure and other phenomena which occur in hydraulic systems. It is suggested that the authors, in their final analysis of friction characteristics of O-rings, prepare a technical chart wherefrom hydraulic designers can pick friction information for their particular designs at a glance.

Authors' Closure

The authors wish to thank the critics for their remarks about the work. Mr. Davisson's discussion of the background of the project is a fitting addition to this summary of facts. Many of the questions raised in the discussion are beyond the narrow scope of this paper, but are welcomed because they propose a rather logical course for further investigation. They introduce problems of fundamental importance to the hydraulics industry.

For all data reported, rod travel was in a direction opposite to the direction of applied working pressure. This is almost always the performance condition for rod seals when an hydraulic cylinder is operated. However, limited tests were made for the reverse case in which travel was in the direction of the applied working pressure. A "toe-stubbing" or high-frequency "stick-slip" type of friction was observed which was considered unworthy of analysis. Recent endurance-cycling tests at Battelle indicate that this application shortens O-ring life.

This study was confined to room temperatures which averaged about 90 F, but actually ranged from 60 to 115 F. This was dependent somewhat upon operating conditions, but mostly upon the season of the year. This factor is considered to be only a minor cause for the rather wide deviations reported. Inspection of the data and temperatures at which single measurements were made seems to indicate that running and breakout friction increase slightly with temperature. This was not demonstrated conclusively, however.

---

[6] Professor of Mechanical Engineering, Iowa State College of Agriculture and Mechanic Arts, Ames, Iowa. Mem. ASME.

[7] Project Engineer, Aircraft Development, Weatherhead Company, Cleveland, Ohio. Mem. ASME.

In so far as packing life at higher temperatures is concerned, it is well known that rubber ages much more rapdly at higher temperatures. Consequently, an appreciably shorter packing life would be expected for an application involving temperatures in excess of that for which the packing was designed. AN-P-79 packings are designed for a maximum temperature of 160 F. Many commercial-type packings have been designed for still higher temperatures. An upper limit for rubber-type packings is believed to be in the range of 250 F.

Zero-pressure breakout friction was not included in the work reported here. However, a study of certain unreported data has shown that zero-pressure breakout may be determined by extrapolation to zero pressure of the values determined at 500, 1000, and 1500 psi, when the other factors remain constant. This method generally determines values within the deviation limits cited, but they are occasionally high by as much as 50 per cent.

Time delays in excess of 40 min., and O-ring installations with leather back-up rings were not studied. Early work at Battelle showed that time delays of several days gave only slightly greater breakout friction than that obtained for 40 min. The maximum delays reported are, therefore, approximately representative of the maximum breakout developed by O-rings during pressurized stand-by periods. Over a long period of time, for example, several months in a service installation, other factors such as permanent set and fouling of the extended rod, due to heat and dust exposure, would be expected to increase breakout friction a significant amount.

Variable acceleration, if it existed, was dependent upon stroke speed, due to the nature of the test machine. Therefore, it is believed legitimate to express breakout friction as a fuction of stroke speed in Fig. 7 of the paper. Recent measurements have shown that rod acceleration was in the order of 100 fps per sec. Therefore, a full-stroke speed of 300 ipm was reached within about 0.004 sec after start. Recent oscillograph-type measurements of friction, which correlate very well with the engine indicator-type measurements of friction reported in this paper, show that breakaway occurs during this acceleration period.

The mass of the bellows and engine indicator are believed to have introduced some inertia factor into the breakout results reported. However, very little displacement of these components during breakout was observed, making this factor a negligible one, in light of the deviations reported. The authors realize the need for a method of measuring breakout friction which would be totally independent of any inertia interference.

The initial position of the O-ring recently has been observed to vary unpredictably during the stand-by period. An initial static-friction pull on the rod due to pressurization of the O-ring alone has been seen to vary from zero to 5 lb. Considering the high-order acceleration involved during breakaway, this is believed to account for the large deviations in the breakout-friction results. This irreproducibility would, however, be characteristic of actual O-ring installations in hydraulic systems.

It was seen that running and breakout friction decreased with increased rod smoothness for the range of surface finishes studied. Although two points at 1.7–2.0 microin. rms in Fig. 5 of the paper, are out of line and indicate higher friction, a different O-ring was tested for this surface finish. The reproducibility of friction between different O-rings is illustrated in Fig. 8, and could account for the deviations from the smooth curves drawn in Fig. 5.

Since the presentation of this paper, considerable effort has been directed toward determining the effect of O-ring size on friction, and correlating the accumulated data into technical design charts. In general, it was found that the smaller sizes developed less friction, as expected. The resultant data, however, could not be fitted into a correlation which could be extropolated to all sizes. This was due to the large variability in our data, and also to many dimensional factors which were difficult to control closely. Running-friction characteristics were much more consistent than breakout friction, and appear capable of being resolved into such charts. More sensitive test methods will be required to do this, however.

It cannot be said that the present AN specifications represent the optimum in O-ring material. Instead, they represent the minimum requirements, which would enable United States military aircraft to operate anywhere in the world with the same hydraulic fluid and packings. In the development of materials to meet this broad objective, many compromises in physical properties had to be made. Oil resistance, for example, was sacrificed for low-temperature flexibility. O-rings with physical properties superior to AN compounds have been formulated for individual industrial applications, where low-temperature flexibility is not a problem and wider latitude exists in the choice of a hydraulic fluid.

The frictional characteristics of O-rings are believed indicative of their expected life in service, within limits. High breakout friction is evidence of extrusion, where the packing is more susceptible to "nipping" when the rod is actuated. O-ring packings will tend to wear away faster by abrasion when running friction is high. Endurance cycling tests show, however, that the wearing rate of the packings does not correlate strictly with O-ring life, though wear is proportional to the number of completed strokes. Further investigation revealed that strict definition and control of pressure-time characteristics is of utmost necessity in conducting life tests, and being able to predict how long the O-rings will last in service. Accurate measurement and close control of the pressure-time relations in hydraulic systems is a serious problem in itself. This will have to be solved before the effect of the finite packing properties on O-ring life can be determined.

TABLE 3 SUMMARY OF GLAND-DESIGN FACTORS

	Low pressure (500 psi)		High pressure (1500 psi)	
Type of application	Squeeze	Surface finish	Squeeze	Surface finish
Intermittent cycling (breakout friction is important)	Maintain as low as possible	Maintain as smooth as possible	Unimportant 10 to 16%	Unimportant in 2- to 11-microin. range
Continuous cycling (running friction is important)	Maintain as low as possible	Maintain as smooth as possible	Unimportant 10 to 16%	Maintain as smooth as possible

It is believed that the friction studies point out certain gland-design considerations which can be applied to lenghten O-ring life. These are summarized in Table 3 of this closure.

Strength-Variance Studies of Plastics

By W. J. GAILUS,[1] STEVEN YURENKA,[2] AND A. G. H. DIETZ[3]

Preliminary work of the plastics research program at the Massachusetts Institute of Technology, sponsored by the Plastic Materials Manufacturers Association is reported. A statistical study of the effect of pertinent variables likely to be encountered in gathering mechanical data from a representative system of plastics (polymethyl methacrylate) is described. Effect of manufacturer, rate of test, and nature of test specimen is reported.

INTRODUCTION

BASIC to the plastics research program reported in this paper was the need for a clear understanding of the variables likely to be important and for a rather complete knowledge, from a physical viewpoint, of the homogeneity and original "as-received" condition of the materials to be tested. To achieve this goal, it was decided to obtain randomized physical property data on the plastics concerned as part of a preliminary exploratory program. From such information it was hoped that it would be possible to reach a better-informed decision as to the number and distribution of test specimens to be used in the future major work; and consequently, the quantities and type of material required.

Polymethyl methacrylate was chosen as the first material to be studied in detail because it was a representative thermoplastic, was readily available in homogeneous form, and had a comparatively simple structure which could, however, be altered chemically under controlled conditions to give a variety of structural forms. The polymethyl methacrylate was supplied by two manufacturers (hereinafter referred to as A and B).

Since fairly large numbers of specimens of different shapes were to be used in the main program, uniformity as regards the material going into the various types of specimens was of prime importance, and so it was decided to machine them from cast-sheet stock. In order to obtain the requisite information just mentioned, a preliminary statistical program was designed to investigate the following variables:

1 Variation in the material supplied by manufacturers A and B to determine generally over-all differences in the material due to inherent variations, and differences in the manufacturing techniques used by A and B.

2 Variations of the material taken from different positions in a particular sheet—center, corner, center of edge, and intermediate—all with the view in mind of determining the degree of uniformity within any one sheet itself as regards positional variations.

3 Variation of the material taken from sheets of different thicknesses, $\frac{1}{4}$ in., $\frac{1}{2}$ in., and 1 in. to obtain an indication of the uniformity in the sheets as regards material cast in different sheet thicknesses.

4 Variation of material taken from the surface of the sheets, as compared with material from the center of the thicknesses, to indicate the uniformity of the material in so far as depth from the surface was concerned.

5 Variation in measured physical properties as influenced by two testing speeds, differing by ratios of approximately 10 to 1. The reason for including item (5) was twofold: (*a*) to get an indication of the importance of the rate of test; and (*b*), to have a basis of relative comparison for the importance of the first four variations. It is the results of this preliminary survey that are presented in this paper.

MATERIAL AND TEST PROCEDURE

The material investigated was supplied in duplicate by manufacturers A and B in the form of cast sheets in three thicknesses, $\frac{1}{4}$ in., $\frac{1}{2}$ in., and 1 in., and approximately 36 in. × 36 in. in size. Manufacturer A supplied straight commercial-run stock while manufacturer B supplied material manufactured in a commercial run but under the supervision of laboratory personnel. In addition, the material supplied by B was given subsequent annealing treatments.

To facilitate that portion of the program concerned with specimen manufacture and of actual testing, it was decided to use simple bending tests throughout this preliminary work to obtain the physical data to be used in the calculations. The specimens employed were 0.2 in. thick, 0.5 in. wide, and of such length that the tests could be carried out with 3.2 in. between the supports (span-depth ratio of 16 to 1). Loads on the specimens were applied at the center of the span. The positions of the various specimens with respect to the sheets are as shown in Fig. 1.

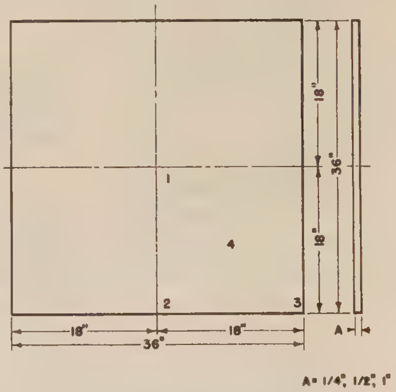


FIG. 1 SAMPLE POSITIONS CHOSEN FROM CAST SHEETS OF POLYMETHYL METHACRYLATE

The numbered positions are diagrammatic, i.e., the group of specimens from position 3, for example, might have been taken from any of the four corners, not necessarily the one shown. Surface specimens were machined from the same surface of the sheet; interior specimens were machined from the geometric center plane of the sheets. Since the major portion of the samples to be used in the main testing program would have machined surfaces,

---

[1] Research Associate, Plastics Research Laboratory, Department of Building Engineering and Construction, Massachusetts Institute of Technology, Cambridge, Mass. Jun. ASME.

[2] Research Associate, Plastics Research Laboratory, Department of Building Engineering and Construction, Massachusetts Institute of Technology.

[3] Associate Professor of Structural Engineering; Director, Plastics Research Laboratory, Department of Building Engineering and Construction, Massachusetts Institute of Technology, Mem. ASME.

Contributed by the Rubber and Plastics Division and presented at the Fall Meeting, Erie, Pa., September 28–30, 1949, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 49—F-33.

it was felt desirable to minimize the complication of the "skin effect;" therefore the surface specimens were loaded with the molded skin on the upper, or compression, side. The tests were conducted in an air-conditioned atmosphere of 77 F and 50 per cent relative humidity (RH) at constant rates of crosshead motion (effectively constant rates of deflection) at speeds of 0.10 and 1.00 inches per minute.

ANALYSIS PROCEDURE

The value of an experiment is measured directly by the reliability of the measurements made. Reliable measurements in turn depend upon complete, or at least adequate, recognition of the variables involved and of their relative importance. The analysis of variance assigns to each of the five previously mentioned experimental variables (and combinations of them) its appropriate share of the total experimental variation and then tests whether each such share is significantly high or merely due to chance. With the aid of the results of such an analysis, any future program can be planned with confidence. If the analysis shows a large variation, for example, due to the position from which the specimen is taken in the sheet, or due to sheet thickness, then, in future work, an effort can be made to take specimens for a given series of tests from positions close to the same spot in a given sheet, or from sheets of the same thickness. In addition to evaluating the main sources of variation, these tests give a measure of the inherent and unavoidable variation in the results to be expected due to the host of uncontrolled variables which are acting, such as slight variations in temperature, humidity, speed of test, and dimensions. This uncontrollable variation or error is used as a base value in determining the number of tests, and so the amount of material, necessary for future testing work. Details on this point will be given in a later section.

In a complete formal analysis of the five variables being considered, there are 31 categories into which the total variation would be divided. These include the five main effects of testing rate, sheet thickness, manufacturer, position in the sheet, and "depth" in the sheet, i.e., surface as compared to internal specimens. These are the most likely sources of significant variation. In addition, there are ten first-order interaction effects such as the interaction of rate and sheet thickness, manufacturer and depth, position and depth, and so forth. The variation due to an interaction term would be a variation in one factor as caused by a variation in another factor. For example, consider the interaction term "testing rate and depth." It may very well be that the total average of maximum strengths would be the same for surface specimens as for internal specimens, yet the fast surface specimens may test significantly high compared to the fast internal specimens with the slow surface specimens significantly low compared to the slow internal specimens. The magnitude of the first-order interaction term, rate and depth, would indicate this condition. In addition to the fifteen variational terms already mentioned, there are ten second-order interaction terms such as "rate × depth × manufacturer," and "sheet thickness × rate × position." Further, there are also five third-order interaction terms, and one fourth-order term. These higher-order terms are of the same nature as the first-order interactions described, but much more difficult to comprehend and to associate with a physical meaning.

CALCULATION MECHANICS

The entire following discussion will be concerned with the maximum stress values obtained from the bending tests using the formula $s = Mc/I$. A completely separate and similar analysis must be made if the modulus of elasticity values are considered.

After the tests have been completed and the data tabulated

TABLE 1 MAXIMUM BENDING STRESS (Mc/I) PSI; TESTS AT CONSTANT RATE OF CROSSHEAD MOTION

	Material A			Material B		
	Original Sheet Thickness			Original Sheet Thickness		
	¼"	½"	1"	¼"	½"	1"
Surface Specimens						
Fast Rate Tests						
Position 1	17,400	19,100	19,130	15,000	16,900	20,100
" 2	19,900	18,500	15,870	13,500	17,400	20,200
" 3	17,300	18,600	17,520	15,050	19,500	17,860
" 4	20,200	21,200	20,940	14,300	16,800	16,320
Slow Rate Tests						
Position 1	18,150	16,100	17,500	13,500	15,100	17,400
" 2	19,300	18,000	18,490	12,250	16,400	16,800
" 3	16,400	16,000	15,280	13,200	13,600	17,700
" 4	18,900	17,450	17,610	13,400	13,200	16,900
Internal Specimens						
Fast Rate Tests						
Position 1	18,600	22,300	9,990	15,800	18,800	20,200
" 2	19,900	22,400	13,760	14,850	16,500	11,960
" 3	14,200	16,500	16,320	15,450	16,100	16,320
" 4	20,450	22,600	13,400	14,100	15,400	16,820
Slow Rate Tests						
Position 1	16,800	18,500	9,640	14,100	14,850	16,700
" 2	18,000	20,600	15,190	13,800	15,100	16,800
" 3	17,500	18,950	12,820	11,850	16,200	17,800
" 4	18,050	19,400	13,790	13,400	15,800	16,600

TABLE 2 MODIFIED MAXIMUM BENDING-STRESS VALUES
(Actual stress less 10,000 and divided by 10)

Position	Material	Original Sheet Thickness			Total
		$\frac{1}{4}$"	$\frac{1}{2}$"	1"	
Slow Rate, Surface Specimens					
1	A	815	610	750	2,175
	B	350	510	740	1,600
2	A	930	800	849	2,579
	B	225	640	680	1,545
3	A	640	600	528	1,768
	B	320	360	770	1,450
4	A	890	745	761	2,396
	B	340	320	690	1,350
Sub Totals	A	3275	2755	2888	8,918
	B	1235	1830	2880	5,945
Total		4510	4585	5768	14,863
Fast Rate, Surface Specimens					
1	A	740	910	913	2,563
	B	500	690	1010	2,200
2	A	990	850	587	2,427
	B	350	740	1020	2,110
3	A	730	860	752	2,342
	B	505	950	786	2,241
4	A	1020	1120	1094	3,234
	B	430	680	632	1,742
Sub Totals	A	3480	3740	3346	10,566
	B	1785	3060	3448	8,293
Total		5265	6800	6794	18,859

TABLE 3 MODIFIED MAXIMUM BENDING-STRESS VALUES
(Actual stress less 10,000 and divided by 10)

Position	Material	Original Sheet Thickness			Total
		$\frac{1}{4}$"	$\frac{1}{8}$"	1"	
Slow Rate, Internal Specimens					
1	A	680	850	-36	1,494
	B	410	485	670	1,565
2	A	800	1060	519	2,379
	B	380	510	680	1,570
3	A	750	895	282	1,927
	B	185	620	780	1,585
4	A	805	940	379	2,124
	B	340	580	660	1,580
Sub Totals	A	3035	3745	1144	7,924
	B	1315	2195	2790	6,300
Total		4350	5940	3934	14,224
Fast Rate, Internal Specimens					
1	A	860	1230	-11	2,079
	B	580	880	1020	2,480
2	A	990	1240	376	2,606
	B	485	650	196	1,331
3	A	420	650	632	1,702
	B	545	610	632	1,787
4	A	1045	1260	340	2,645
	B	410	540	682	1,632
Sub Totals	A	3315	4380	1337	9,032
	B	2020	2680	2530	7,230
Total		5335	7060	3867	16,262

(Table 1), the information may be entered for analysis in the form shown in Tables 2 and 3. The original maximum stress values from Table 1 have been decreased by 10,000 and divided by 10 to reduce the size of the terms to be encountered in later arithmetical manipulations. (In general, all values in the initial data may be multiplied by a constant, or have a constant added to them without affecting the results in any way.)

The entire original data have been plotted in the frequency-distribution bar graph, Fig. 2. It can be seen that the maximum bending stresses range between 9000 and 22,000 psi, and that the arithmetic mean of the stresses for materials A and B and for their total lie between 15,000 and 17,000 psi. The ordinate of this graph is calibrated in frequency units or the number of specimens which broke within the range of stresses represented along the abscissa.

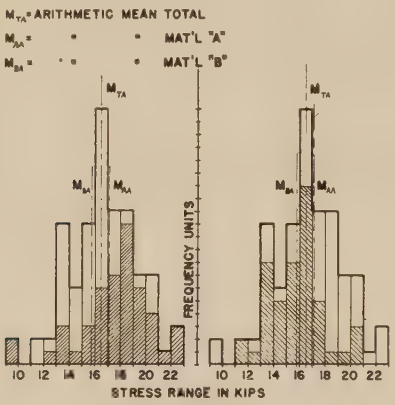


Fig. 2 Variance Series Frequency Distribution; Maximum Stress in Bending

Below each table the sum of values for each thickness and manufacturer is formed, and horizontally, the sum of values for each position and manufacturer is similarly obtained. This is done to facilitate construction of further tables, as will presently be appreciated. The sum of all values for a given manufacturer may be obtained by a summation of either the horizontal or vertical sums. This latter operation provides a check on the preceding computations.

Tabulations like the ones shown in Tables 4, 5, and 6 (there would be ten such tables if all were listed, one for each of the first-order interactions) are then filled in with the appropriate sums. The totals which were formed in Tables 2 and 3 are now found to be convenient aids for making all the entries into Tables 4, 5, 6, etc., except in the case of the position versus thickness table (Table 6). Here, for example, the first entry is obtained by using the individual values in Tables 2 and 3, or 815 + 350 + 740 + 500 + 680 + 410 + 860 + 580 = 4935. Considering now for a moment Table 4, it may be seen that the first entry is the sum of all fast-rate, "A" specimens, or 10,566 (from Table 2) + 9032 (from Table 3) = 19,598. In a similar manner the remaining three entires are obtained. As a further example, consider Table 5. The first entry is the sum of all fast-rate, position 1 specimens or 2563 + 2200 (both from Table 2) + 2079 + 2480 (both from Table 3) = 9322. This procedure is followed in filling in the remaining values.

Finally, Table 7 may be made up, putting the "sum of squares of the deviation" in the first column.

The sum of squares of deviation is defined by the series

$$\sum_{1}^{n} (\bar{x} - \bar{\bar{x}})^2$$

where $\bar{x}$ is the mean value of maximum stress considering a particular variable and $\bar{\bar{x}}$ is the mean value of maximum stress considering all of the data; n is the total number of readings or values considered. To clarify this situation a bit, take, for example, the case of the interaction "position and thickness" (see Table 6). The discussion will be simplified considerably by introducing here a few symbolisms. Let

Σx = total sum of all maximum stress values = 64,208

Σx_P = total sum of maximum stress values for any one position

= 16,156 for position 1, etc. (four values in all)

Σx_T = total sum of maximum stress values for any one thickness

TABLE 4 VARIANCE COMPUTING TABLE; RATE OF TEST VERSUS MANUFACTURER

Manufacturer	Fast Rate	Slow Rate	Total
A	19,598	16,842	36,440
B	15,523	12,245	27,768
Total	35,121	29,087	64,208

(0) Sub-total sum of squares $= \frac{1}{n_{RM}} \sum_{1}^{4} (x_{RM})^2 = \frac{1}{24} \left[(19{,}598)^2 + \ldots \right] =$ 44,109,921.7

(1) Sum of squares of Rate Deviations $= \sum_{1}^{n} (\bar{x}_R - \bar{\bar{x}})^2 = \frac{1}{48} \left[(35{,}121)^2 + (29{,}087)^2 \right] - \frac{1}{96} (64{,}208)^2 = 379{,}262.1$

(2) Sum of squares of Manufacturers Deviations $= \sum_{1}^{n} (\bar{x}_M - \bar{\bar{x}})^2 =$ $\frac{1}{48} \left[(36{,}440)^2 + (27{,}768)^2 \right] - \frac{1}{96} (64{,}208)^2 = 783{,}370.7$

(3) Interaction of Rate vs. Manufacturer $= \sum_{1}^{n} (\bar{x}_{RM} - \bar{x}_R - \bar{x}_M + \bar{\bar{x}}) = 2838.3$

TABLE 5 VARIANCE COMPUTING TABLE; POSITION VERSUS RATE OF TEST

Position	Fast Rate	Slow Rate	Total
1	9,322	6,834	16,156
2	8,474	8,073	16,547
3	8,072	6,830	14,802
4	9,253	7,450	16,703
Total	35,121	29,087	64,208

(0) Sub-total sum of squares $= \frac{1}{n_{PR}}\sum_{1}^{8}(x_{PR})^2 = \frac{1}{12}\left[(9{,}322)^2 + \ldots\right] =$ 45,121,653.2

(1) Sum of squares of Rate Deviation $= \sum_{1}^{n}(\bar{x}_R - \bar{\bar{x}})^2 = \frac{1}{48}\left[(35{,}121)^2 + (29{,}087)^2\right] - \frac{1}{96}(64{,}208)^2 = 379{,}262.1$

(2) Sum of squares of Position Deviation $= \sum_{1}^{n}(\bar{x}_P - \bar{\bar{x}})^2 = \frac{1}{24}\left[(16{,}156)^2 + \ldots\right] - \frac{1}{96}(64{,}208)^2 = 93{,}422.6$

(3) Interaction of Position vs. Rate $= \sum_{1}^{n}(\bar{x}_{PR} - \bar{x}_P - \bar{x}_R + \bar{\bar{x}}) = 1{,}704{,}517.9$

TABLE 6 VARIANCE COMPUTING TABLE; POSITION VERSUS THICKNESS

	Original Sheet Thickness			
Position	¼"	½"	1"	Total
1	4,935	6,165	5,056	16,156
2	5,150	6,490	4,907	16,547
3	4,095	5,545	5,162	14,802
4	5,280	6,185	5,238	16,703
Total	19,460	24,385	20,363	64,208

(0) Sub-total sum of squares $= \frac{1}{n_{PT}}\sum_{1}^{12}(x_{PT})^2 = \frac{1}{8}\left[(4{,}935)^2 + (5{,}150)^2 + (4{,}095)^2 + \ldots\right] = 43{,}547{,}174.7$

(1) Sum of squares of Thickness Deviation $= \sum_{1}^{n}(\bar{x}_T - \bar{\bar{x}})^2 = \frac{1}{n_T}\sum_{1}^{3}(x_T)^2 - \frac{1}{n}\left(\sum_{1}^{96}x\right)^2 = \frac{1}{32}\left[(19{,}460)^2 + \ldots\right] - \frac{1}{96}(64{,}208)^2 =$ $43{,}374{,}112.3 - 42{,}944{,}450.6 = 429{,}661.7$

(2) Sum of squares of Position Deviation $= \sum_{1}^{n}(\bar{x}_P - \bar{\bar{x}})^2 = \frac{1}{24}\left[(16{,}156)^2 + \ldots\right] - \frac{1}{96}(64{,}208)^2 = 43{,}037{,}937.2 - 42{,}944{,}450.6 =$ 93,422.6

(3) Interaction of Position vs. Thickness $= \sum_{1}^{n}(\bar{x}_{PT} - \bar{x}_P - \bar{x}_T + \bar{\bar{x}})^2 = 79{,}575.8$

TABLE 7 OVER-ALL VARIANCE ANALYSIS

Cause	Sum of Squares of Deviation	Degrees of Freedom	Mean Square Deviation	$\frac{\text{M.S.D.}}{3490}$
1) Manufacturer	783,370.7	1	783,370.7	224.46
2) Rate of Test	379,262.1	1	379,262.1	108.67
3) Depth	109,080.2	1	109,080.2	31.26
4) Thickness	429,661.7	2	214,330.8	61.56
5) Position	93,422.6	3	31,140.9	8.92
6) Rate x Manufacturer	2,838.3	1	2,838.3	0.81
7) Depth x Manufacturer	34,504.1	1	34,504.1	0.99
8) Thickness x Manufacturer	1,645,879.8	2	822,939.9	235.80
9) Position x Manufacturer	435,061.9	3	145,020.6	41.53
10) Position x Depth	757.6	3	252.5	0.07
11) Rate x Depth	258,095.4	1	258,095.4	73.95
12) Thickness x Depth	927,902.5	2	463,951.3	132.94
13) Rate x Position	1,704,517.9	3	568,172.6	162.80
14) Thickness x Position	79,575.8	6	13,262.3	3.80
15) Thickness x Rate	91,670.0	2	45,835.0	13.13
16) 2nd 3rd 4th Order Interactions	219,868.9	63	3,490.0	1.00
17) TOTAL	7,195,535.4	95	-----	----

= 19,460 for the $\frac{1}{4}$ in. thickness, etc. (three values in all)

Σx_{PT} = total sum of strength readings for any one combination of a specific thickness and a specific position = 4935 for position 1, $\frac{1}{4}$ in. thickness, etc. (twelve values in all)

n_P = number of readings totaled to form each Σx_P $= \frac{\text{total number of values}}{\text{number of sums } P} = \frac{96}{4} = 24$

n_T = number of readings totaled to form each Σx_T $= \frac{\text{total number of values}}{\text{number of sums } T} = \frac{96}{3} = 32$

n_{PT} = number of readings totaled to form each Σx_{PT} $= \frac{\text{total number of values}}{\text{number of sums } PT} = \frac{96}{12} = 8$

n = total number of values = 96

$i = \frac{n_T}{n_{PT}}$ = number of positions studied = 4

$j = \frac{n_P}{n_{PT}}$ = number of thicknesses studied = 3

The sum of squares of deviation between the mean maximum stress values for each thickness and the mean maximum stress for all values may now be written in the form

$$\sum_1^n (\bar{x}_T - \bar{x})^2 = \frac{1}{n_T}\sum_1^j (\Sigma x_T)^2 - \frac{1}{n}(\Sigma x)^2 \ldots\ldots [1]$$

The equation for computing the position deviation is analogous and of the form

$$\sum_1^n (\bar{x}_P - \bar{x})^2 = \frac{1}{n_P}\sum_1^i (\Sigma x_P)^2 - \frac{1}{n}(\Sigma x)^2 \ldots\ldots [2]$$

Finally, the equation for computing the deviation due to the interaction of position and thickness may be written in the form

$$\sum_1^n (\bar{x}_{PT} - \bar{x}_P - \bar{x}_T + \bar{x})^2 = \frac{1}{n_{PT}}\sum_1^{i\cdot j} (\Sigma x_{PT})^2 - \frac{1}{n_T}\sum_1^j (\Sigma x_T)^2 - \frac{1}{n_P}\sum_1^i (\Sigma x_P)^2 + \frac{1}{n}(\Sigma x)^2 \ldots\ldots [3]$$

Similar equations can be written for all other variables and interactions. The calculations involved in evaluating these equations numerically are shown for compactness, below each of the Tables 4, 5, 6, etc. as Equations [1], [2], and [3].

The procedure just described is followed for all entries in Table 7, including the total values, with the exception of the term consisting of the second-, third-, and fourth-order interactions. This term is equal to the total immediately below it minus all the terms above it.

The manner in which the foregoing procedure has been presented requires that the effects of the second-, third-, and fourth-order interactions be lumped together in one sum. To separate out the second-order interaction would require a duplication of about as much effort as has already been expended in making up the present tables. A new sheet with ten additional three-way instead of two-way subtables would be required.

The degrees-of-freedom items of the second column of figures in Table 7 are obtained by subtracting one unit from the total number of items in each category. Thus, for example, because there are two manufacturers, and one of these may be "assumed," the degrees of freedom for the manufacturers is the remainder, or one. Also, since three different thicknesses of specimens were tested, and one of these may be assumed, the number of degrees of freedom is the remainder, or two, etc. Since the total number of specimens tested was 96, the total degrees of freedom is one less or 95.

The mean-square deviation, or variance, listed in the third column of Table 7, is obtained by dividing each sum of squares of deviation by its respective degrees-of-freedom value. Each of these mean-square deviations is an estimate of the variance of the 96 maximum stresses. If all 96 values were "alike," i.e., differed only at random, and had no significant trend between manufacturers, rates, thicknesses, etc., then all the estimates of variance would be alike. By alike is meant here that the values will differ from each other in magnitude, but not significantly.

Actually, from the values of the mean-square deviations shown in Table 7 we can conclude in the case of the first five causes that there was a significant difference in strength between materials A and B, since the estimate of variance due to manufacturers is significantly high. Furthermore, the estimate of variance due to position is significantly low compared to the values of the other four causes, which would indicate that there was not much difference in the maximum strength of specimens taken from various positions of the test plate or, in other words, the test plate is fairly homogeneous. It is important here to emphasize the fact that this conclusion is a relative one based upon consideration of the five primary causes, i.e., of these five causes, the position variation is the smallest. This is all that can be said at this point. The causes taken in order of importance of their influence on the maximum stress values obtained are as follows: (1) manufacturer, (2) rate of test, (3) thickness, (4) depth, (5) position.

In the case of the interaction terms, the analysis is not so simple, but here again it can be seen that the greatest influence on the maximum stress is that of cause (8), thickness versus manufacturer; and the least influence is that of cause (10) or position versus depth. The higher-order interaction terms (second, third, and fourth order in our case) do have physical meanings especially to engineers. Statisticians rarely include such in their analyses and indeed we have already obtained all the information we want so that the second and higher interactions are combined and taken as the variance or variability of the material due to such other variables not specifically considered here, such as fluctuations in temperature and humidity, and random errors in measuring dimensions, rates, deformations, etc. This variance is, so to speak, the basic, unavoidable, and unallocatable variation; and is used as the basis for comparison by which the other variances are judged significant.

The mean-square deviations which are smaller than the second-, third-, and fourth-order interactions combined obviously are not significant, but those which are considerably larger should be analyzed. To facilitate comparison, a convenient tabulation is obtained by dividing the mean-square deviation of the effect being tested by the mean-square deviation due to second-, third-, and fourth-order interactions (the lumped variation). The values obtained are shown in the last column of Table 7 and are further plotted in the form of a bar graph in Fig. 3. It is readily apparent that the general effect of manufacturer is greatly pronounced, overshadowing all else except thickness × manufacturer, which might conceivably also be due to the manufacturer participation. The rate of test is shown to be highly significant also, as might be expected. Thickness × depth is certainly significant and would indicate the necessity of expecting different results from tests on specimens taken from different thicknesses at varying depths below the surface even when the tests are somewhat identical. The same is true for rate × position.

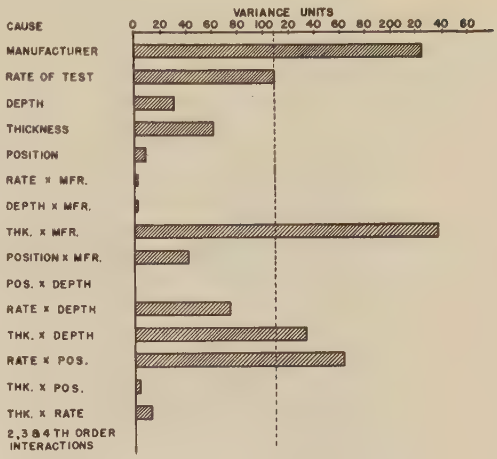


FIG. 3 VARIANCE SERIES COMPARATIVE VARIANCES

SUBDIVIDED ANALYSIS

The over-all statistical studies here reported have grouped interaction terms which hinder specific allocation of effect to their proper factors. Thus to get a clearer idea of the relative magnitudes of the variations due to the various causes, a further, subdivided variance study has been made. Analyses of materials A and B separately to eliminate cross-terms attributable to variations between manufacturing techniques were thought most likely to clarify the situation. The computations made were carried out in the manner previously explained in detail, and have been summarized in Tables 8, 9, 10, and 11. The mean-square deviations in the last column of Tables 9 and 11 have also been plotted in the form of a bar graph in Fig. 4. Since the maximum order of magnitude was three, the analysis was carried out in detail to study the third-order effects, as shown in the tables and graphs.

Both of these subdivided analyses indicate that the largest deviation is caused by differences in physical properties of sheets of different thicknesses of apparently identical material. The remaining "mean-square-deviation" terms of the material B analysis seem to indicate that the effect of rate of loading is sizable, as normally would be expected, and, further, that positional variations and depth variations (surface versus internal

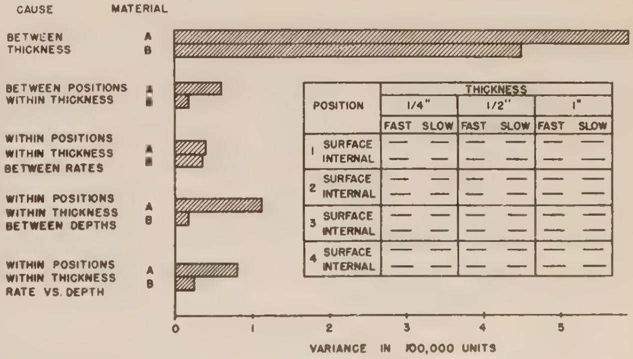


FIG. 4 VARIANCE SERIES COMPARATIVE VARIANCES

TABLE 8 VARIANCE STUDY; MATERIAL B COMPUTATIONS

Position		Thickness								
		1/4"			1/2"			1"		
		Fast	Slow	Total	Fast	Slow	Total	Fast	Slow	Total
1	Surface	500	350	850	690	510	1200	1010	740	1750
	Internal	580	410	990	880	485	1365	1020	670	1690
	TOTAL	1080	760	1840	1570	995	2565	2030	1410	3440
2	Surface	350	225	575	740	640	1380	1020	680	1700
	Internal	485	380	865	650	510	1160	196	680	876
	TOTAL	835	605	1440	1390	1150	2540	1216	1360	2576
3	Surface	505	320	825	950	360	1310	786	770	1556
	Internal	545	185	730	610	620	1230	632	780	1412
	TOTAL	1050	505	1555	1560	980	2540	1418	1550	2968
4	Surface	430	340	770	680	320	1000	632	690	1322
	Internal	410	340	750	540	580	1120	682	660	1342
	TOTAL	840	680	1520	1220	900	2120	1314	1350	2664

TABLE 9 VARIANCE STUDY; MATERIAL B ANALYSIS

Effect	Sum of Squares of Deviation	Degrees of Freedom	Mean Square Deviation
1) Between Thicknesses	899,784	2	449,892
2) Between Positions within Thicknesses	171,365	9	19,040.6
3) Within Positions within Thicknesses Between Rates	432,153	12	36,012.8
4) Within Positions within Thicknesses Between Depths	228,315	12	19,026.3
5) Within Positions within Thicknesses Rate vs. Depth	319,628	12	26,635.7
6) TOTAL	2,051,245	47	

specimens) are of the same order of importance, namely, about one half the magnitude of the deviation due to rate of loading, indicating fairly homogeneous material.

In the material A analysis, it is to be noted that the deviation "between thicknesses" is very high, indicating variability in the material taken from different sheets is significant. The analysis also indicates that deviation "between positions" is also high, indicating nonuniformity over a single sheet. Further, "depth" variation is also significant, indicating a nonhomogeneous structure in material near the surface of the sheet, as compared with material near the center plane.

CONCLUSION

This variance analysis of causes and interaction terms indicates that studies correlating structural characteristics of a material, whether chemical, metallurgical, etc., with its physical properties had best be done by testing samples whose homogeneity and degree of representation (reproducibility) of a material are understood and can, therefore, be taken into consideration. Further, the importance of considering interaction effects has been indicated. The latter, if not considered, can easily lead to false conclusions or effectively mask effects that may occur.

In the case of the plastic materials analyzed in the paper, it has been found that in order to eliminate as many interactions as possible, future testing should be carried out using material (for particular phases of the work) from a single manufacturer, with samples taken from panels of the same thickness, using a predetermined random sampling procedure for obtaining specimens.

TABLE 10 VARIANCE STUDY; MATERIAL A COMPUTATIONS

Position		Thickness								
		1/4"			1/2"			1"		
		Fast	Slow	Total	Fast	Slow	Total	Fast	Slow	Total
1	Surface	740	815	1555	910	610	1520	913	750	1663
	Internal	860	680	1540	1230	850	2080	-11	-36	-47
	TOTAL	1600	1495	3095	2140	1460	3600	902	714	1616
2	Surface	990	930	1920	850	800	1650	587	849	1436
	Internal	990	800	1790	1240	1060	2300	376	519	895
	TOTAL	1980	1730	3710	2090	1860	3950	963	1368	2331
3	Surface	730	640	1370	860	600	1460	752	528	1280
	Internal	420	750	1170	650	895	1545	632	282	914
	TOTAL	1150	1390	2540	1510	1495	3005	1384	810	2194
4	Surface	1020	890	1910	1120	745	1865	1094	761	1855
	Internal	1045	805	1850	1260	940	2200	340	379	719
	TOTAL	2065	1695	3760	2380	1685	4065	1434	1140	2574

TABLE 11 VARIANCE STUDY; MATERIAL A ANALYSIS

Effect	Sum of Squares of Deviation	Degrees of Freedom	Mean Square Deviation
1) Between Thicknesses	1,175,757	2	587,878.5
2) Between Positions within Thicknesses	543,511	9	60,390.1
3) Within Positions within Thicknesses Between Rates	470,464	12	39,206.2
4) Within Positions within Thicknesses Between Depths	1,389,377	12	115,781.4
5) Within Positions within Thicknesses Rate vs. Depth	929,810	12	77,484.2
6) TOTAL	4,508,919	47	

and using material which has been prepared by a carefully controlled technique.

ACKNOWLEDGMENT

The authors wish to express their appreciation especially to Prof. L. C. Young for his invaluable assistance in applying and interpreting the techniques of the statistical approach, and to L. Lipschitz for his assistance in setting up the mechanics of the problem in the initial stages of the work. W. Campbell assisted in some of the tests.

Theoretical Considerations on the Optimum Adjustment of Regulators

By P. HAZEBROEK[1] and B. L. van der WAERDEN[1]

In a companion paper, the authors propose some practical rules for the best possible adjustment of the parameters of automatic regulators with proportional-plus-integral action. The present paper will be concerned with the theoretical foundations of the method developed.

Fundamental Notions

IN this paper, we shall assume that a "single" variable is to be kept at a "constant" value by automatic regulation.

We begin by a short survey of the fundamental notions to be used.

Process and regulator are interconnected in such a way as to form a closed loop which we will call a "regulation circuit," Fig. 1. In a regulation circuit it is appropriate, for the purpose of our theory, to distinguish two interdependent variables as follows:

(*a*) The "regulated variable," which we will denote by x, which should be kept as constant as possible by suitable variations of our second variable.

(*b*) The "regulating flow," which we will denote by q.

We will assume that both x and q are measured from their normal value, which shall be taken as zero points and expressed in dimensionless units; for instance, as fractions of the maximum interval in which they may move.

The restriction shall be imposed that the relationship between x and q is such that the superposition principle is valid, in other words that the sum of two time-dependent disturbances of q (or x) causes a change of x (or q) which is the sum of the changes resulting from the two disturbances taken separately. This restriction has to be made in any general theory of regulation.

If the superposition principle holds good, we can confine ourselves, when studying the response of x (or q), to a "standard disturbance," i.e., a sudden deviation 1 from the normal value 0, remaining equal to 1 for all positive values of t.

We define two fundamental functions:

(*a*) The response function $X_R(t)$ of the "process" with respect to the considered variable x. This is the response of x to a standard disturbance of q, the circuit being open and assumed to be severed at A, between process and regulator, Fig. 1.

(*b*) The response function $Q_R(t)$ of the "regulator" is the response of q to a standard disturbance of x taken with a negative sign, the circuit being open and assumed to be severed at B between regulator and process.

Any time-dependent disturbance can be approximated by a succession of small "step functions" and hence can be represented by a linear combination (sum or integral) of standard disturbances. If the response corresponding to the standard disturbance 1 of q is $X_R(t)$, the response corresponding to an arbitrary disturbance $q(t)$ will be

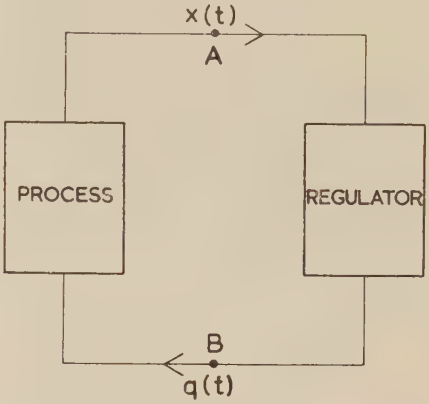


Fig. 1 Diagram of Regulation Circuit

$$\frac{d}{dt}\int_0^t q(t-\tau)X_R(\tau)d\tau \qquad [1]$$

which operation can be conveniently represented by the symbol $q(t) \; o \; X_R(t)$.

In the same way we have for an arbitrary disturbance $x(t)$

$$x(t) \; o \; Q_R(t) = \frac{d}{dt}\int_0^t x(t-\tau)Q_R(\tau)d\tau \qquad [2]$$

Other quantities in the regulation circuit can be derived from $X_R(t)$ and $Q_R(t)$. Especially important is the following function:

Behavior of Regulated Variable $X_B(t)$. This is the response of x to an initial standard disturbance of x under the influence of the regulator. For the sake of brevity this will often be called the behavior function.

The behavior corresponding to an arbitrary time-dependent disturbance can be found by applying a formula of the same kind as Equation [1] or Equation [2] to $X_B(t)$.

Types of Response Curves

In order to study the behavior of the variables in a regulation circuit, it is convenient to assume hypothetical response curves representing mathematical functions distinguished by simple properties. By suitable adjustment to certain parameters occurring in these functions they are adaptable to a more or less close approximation of the majority of response functions met with in practice. The latter point has been treated more fully in the practical part.

For the present we shall confine ourselves to the discussion of the kind of regulation that is performed by a regulator with proportional plus integral action, and which may be represented by the idealized response function

$$Q_R(t) = \begin{cases} 0 & \text{for } t \leqq 0 \\ a + bt & \text{for } t > 0 \end{cases}$$

---

[1] N. V. Bataafsche Petroleum Maatschappij, The Hague, Holland.

Presented at the ASME Industrial Instruments and Regulators Division - Instrument Society of America Conference at St. Louis, Mo., September 12–16, 1949.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 49—IIRD-1.

As for the process, two classes of response functions will be considered as follows:

1 Processes with self-regulation of the following three types:

Type A. Curves having contact of the order $n - 1$ with the axis of t

$$X_R(t) = 1 - e^{-t}\left(1 + t + \frac{t^2}{2!} + \ldots + \frac{t^{n-1}}{(n-1)!}\right) \ldots [3]$$

Type B. Curves with dead time T_d

$$X_R(t) = \begin{cases} 0 & \text{for } t \leqq T_d \\ 1\text{-exp}\left(-\frac{t - T_d}{T_e}\right) & \text{for } t > T_d \end{cases} \ldots\ldots [4]$$

Type C. Abrupt response with dead time T_d

$$X_r(t) = \begin{cases} 0 \text{ for } t \leqq T_d \\ 1 \text{ for } t > T_d \end{cases} \ldots\ldots\ldots\ldots\ldots [5]$$

Type C is a limiting case of Type A ($n \rightarrow \infty$) and also of Type B ($T_e \rightarrow \infty$).

2 Processes without self-regulation of the following two types:

Type D.

$$X_R(t) = t - n + e^{-t}\left[n + (n-1)t + \frac{n-2}{2!}t^2 + \ldots + \frac{1}{(n-1)!}t^{n-1}\right] \ldots\ldots\ldots [6]$$

This function is obtained from Type A by integration from 0 to t.

$$X_R(t) = \begin{cases} 0 & \text{for } t \leqq T_d \\ t - T_d & \text{for } t > T_d \end{cases} \ldots\ldots\ldots\ldots [7]$$

Type E is a limiting case of Type B which may be obtained from the latter by multiplying Equation [4] by T_e and letting T_e become infinite.

As we have pointed out in the practical part,[2] the regulation has to satisfy different requirements according to the asymptotic behavior of the disturbance. In the case of processes with self-regulation, we have taken the standard disturbance as our fundamental disturbance. In the case of processes without self-regulation, we have considered as a fundamental disturbance the time integral of the unit disturbance

$$x(t) = t \quad (t > 0)$$

The behavior of the regulated variable resulting from this disturbance will be called the "second behavior function" and denoted by $X^*_B(t)$. It is the time integral of $X_B(t)$

$$X^*_B(t) = \int_0^t X_B(\tau)d\tau$$

In all these cases the main question is: What choice of the parameters a and b of the regulator gives the best possible regulation, and between what limits can a and b vary without making the regulation much worse?

Criterion for Best Regulation

The new feature of our theory is the putting forward of a criterion for best regulation, which has been proposed by Prof. W. J. D. van Dijck. According to this criterion we shall have optimum performance when the parameters of the response function $Q_R(t)$ (in our case a and b) are given the values for which (1) in the case of processes with self-regulation, the integral

$$S = \int_0^\infty X_B{}^2 dt \ldots\ldots\ldots\ldots\ldots\ldots [8]$$

(2) in the case of processes without self-regulation, the integral

$$S = \int_0^\infty X^*_B{}^2\, dt \ldots\ldots\ldots\ldots\ldots\ldots [8a]$$

attains its minimum value. This is a reasonable condition as it meets both the requirements for good regulation:

(a) The regulated variable should be quickly reduced to its normal value.

(b) The fluctuations of this variable should remain small.[3]

These criteria seem to be especially adapted to the special types of fundamental disturbances we have assumed. We shall prove, however, that it is also satisfactory for more general disturbances.

We will first consider processes with self-regulation.

Let X_B be the behavior of x upon a standard disturbance of x, as previously calculated. The behavior due to an arbitrary disturbance X_D is

$$x = X_D \, o \, X_B = \frac{d}{dt}\int_0^t X_D(t-\tau)X_B(\tau)d\tau \ldots\ldots [9]$$

Now suppose that X_B becomes practically zero after a certain time T, and that X_D is a continuous, differentiable bounded function of t, which means that no sudden, but only gradually increasing and decreasing disturbances occur. In this case Equation [9] may be written

$$x(t) = \int_0^T X_D'(t-\tau)X_B(\tau)d\tau$$

Now, by a well-known inequality of Schwarz

$$\left[\int_a^b f(x)g(x)\,dx\right]^2 \leqq \int_a^b f^2(x)\,dx \cdot \int_a^b g^2(x)\,dx$$

we have

$$x^2(t) = \int_0^T X_D'^2(t-\tau)d\tau \cdot \int_0^T X_B{}^2(\tau)d\tau$$

In general, we may assume that the disturbances met with in practice do not occur so suddenly and so frequently and intensely that the first factor on the right becomes extremely large. As to the second factor, we can make it as small as possible by the application of our criterion. Now if the first factor is not too large and the second one is made small, the whole product will become small, that is, the deviation $x(t)$ will be small for all values of t, which is the object of regulation. Furthermore, if during some time T no disturbances occur, $x(t)$ will be zero after this lapse of time. This is all that reasonably may be expected.

Next we turn to processes without self-regulation. If we make use of the integral, Equation [8], also for these processes, it is found that b becomes zero. When b vanishes, however, the regulated variable would not revert to its normal value when the disturbance increases linearly. As we assume that in the present case the disturbance is of the latter type, the criterion has to be modified. If we replace in our first criterion the integral, Equation [8] by [8a], and if we assume further the disturbance X_D to

---

[2] "The Optimum Adjustment of Regulators," by P. Hazebroek and B. L. van der Waerden, companion paper presented at the IIRD-ISA Conference at St. Louis, Mo., Sept. 12–16, 1949.

[3] Some readers may be inclined to think that our criterion seems somewhat artificial. It might, for instance, be argued that it would be more plausible to choose a and b in such a way that the integral of the absolute value of $X_B(\tau)$

$$S_1 = \int_0^\infty |X_B(\tau)|\,d\tau$$

becomes a minimum. The main objection to this is, however, that this integral is very unsuitable for mathematical treatment. Moreover, the results would in most cases not differ very much from our own.

The same argument is, at least in part, at the bottom of the method of least squares, where the sum of the squares of the deviations is minimized rather than the sum of their absolute values. In fact our method also makes a quantity with the nature of a squared error a minimum.

increase linearly for large t, the foregoing argument is again valid with some slight modifications. The proof may be left to the reader.

Thus the principle of making the integrals, Equations [8] and [8a], respectively, as small as possible for standard disturbances is justified also on the assumption of nonstandard disturbances.

The parameters a and b sometimes must be subjected to certain restrictions, e.g., that the regulated flow should never exceed a prescribed value. In this paper, however, no such restrictions are imposed, and we shall only be concerned with the integral S.

QUALITY OF REGULATION

Let S_m be the minimum of S, corresponding to the best choice of a and b. We introduce an index for the quality of regulation. The quality number N is defined as the quotient

$$N = \frac{S_m}{S}$$

The maximum value $N = 1$ corresponds to the best regulation but regulation with quality number 0.9 is nearly as good.

CALCULATION OF BEHAVIOR OF REGULATED VARIABLE AND REGULATING FLOW

In order to find S and N, it is necessary to calculate the behavior $X_B(t)$ of the variable x.

If at time $t = 0$ a standard disturbance 1 is given to x, the response of q is given by $Q_R(t)$, and the response of x to this regulating flow is given, according to Equation [1], by

$$-F_1(t) = -Q_R(t) \text{ o } X_R(t) = -(Q_R \text{ o } X_R)$$

Hence the regulated variable x does not remain 1, but becomes

$$1 - F_1(t)$$

But this modified disturbance gives rise to a modified response of the regulating flow

$$-(1 - F_1) \text{ o } Q_R$$

and this in turn causes a modified response of the variable x

$$\begin{aligned} -(1 - F_1) \, x \, Q_R \text{ o } X_R &= -(1 - F_1) \text{ o } F_1 \\ &= -F_1 + (F_1 \text{ o } F_2) \end{aligned}$$

Hence the regulated variable becomes

$$1 - F_1 + F_2; \quad (F_2 = F_1 \text{ o } F_1)$$

By continuation of this process we finally find the behavior of x

$$X_B = 1 - F_1 + F_2 - F_3 + \ldots \ldots \ldots \ldots [10]$$

where

$$F_1 = Q_R \, \dot{o} \, X_R = \frac{d}{dt} \int_0^t Q_R(t - \tau) X_R(\tau) d\tau \ldots\ldots [11]$$

$$\left. \begin{aligned} F_2 &= F_1 \text{ o } F_1 \\ F_3 &= F_2 \text{ o } F_1 \end{aligned} \right\} \ldots\ldots\ldots\ldots\ldots [12]$$

In simple cases, for instance, in the case of a step function X_R (Type C), the Equations [10], [11], [12] can be used directly to calculate the behavior X_B. In more complicated cases, other mathematical devices to be discussed later save a considerable amount of labor.

EFFECT OF A DEAD TIME

If one of the response functions Q_R or X_R has a dead time T_d, i.e., remains zero as long as $t \leqq T_d$, the same dead time will appear in F_1, double this time in F_2, treble in F_3, etc. Hence for any finite value of t only a finite number of functions F_n will be different from zero, and the series, Equation [10], has for every finite value of t only a finite number of terms, each of which can be found by direct integration if the functions X_R and Q_R are not too complicated. For large values of t the number of terms may be large, but in that case the whole sum X_B is negligibly small if regulation is at all stable and efficient.

If both functions Q_R and X_R have dead times t_1 and t_2, the sum of these $t_1 + t_2$ will be the dead time of F_1 according to Equation [11]. It is quite irrelevant which part of this sum is due to Q_R and which to X_R; only the sum $t_1 + t_2$ matters. A dead time in Q_R has exactly the same effect as one in X_R; hence we can always assume that Q_R has no dead time. For any satisfactory regulator the dead time has to be very small anyway.

The simplest case for an X_R with dead time is as follows:

Response Process Type C. We can take T_d as unit of time, hence Equation [5] becomes

$$X_R = \begin{cases} 0 \text{ for } t \leqq 1 \\ 1 \text{ for } t > 1 \end{cases}$$

and

$$Q_R = a + bt \quad \text{for} \quad t > 0$$

Equation [11] gives

$$F_1 = Q_R \text{ o } X_R = \begin{cases} 0 & \text{for } t \leqq 1 \\ a + b(t - 1) & \text{for } t > 1 \end{cases}$$

By means of Equation [12], we get

$$F_n(t) = \begin{cases} 0 & \text{for } t \leqq n \\ f_n(t - n) & \text{for } t > n \end{cases}$$

the polynomials $f_1, f_2, \ldots$ being defined by the recursion formulas

$$\begin{cases} f_1 = a + bt \\ f_n = af_{n-1} + b \int_0^t f_{n-1} (\tau) d\tau = \sum\limits_{k=0}^{n} \dfrac{n! \, a^{n-k} b^k}{(n-k)!(k!)^2} \, t^k \end{cases}$$

Finally, Equation [10] gives

$$X_B(t) = 1 - f_1(t - 1) + f_2(t - 2) + \ldots$$

The result of the calculation is shown in Fig. 2, for $a = 0.5$, and $b = 0.8$.

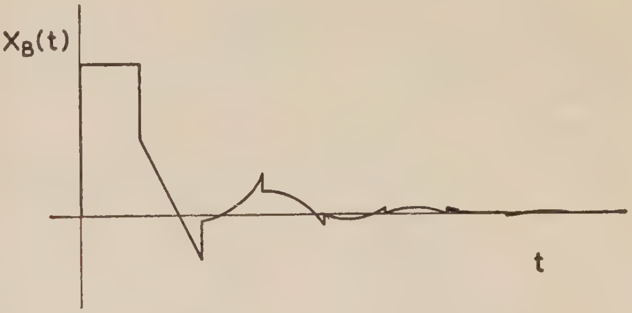


FIG. 2 BEHAVIOR OF REGULATED VARIABLE FOR TYPE C PROCESS

APPLICATION OF OPERATIONAL CALCULUS

In more complicated cases, when direct calculation of the series, Equation [10], is impracticable, the operational calculus is useful. The functions X_R and Q_R give rise to Laplace transforms

$$f(p) = \int_0^\infty pe^{-pt} X_R(t) dt$$

$$g(p) = \int_0^\infty pe^{-pt} Q_R(t) dt$$

The Laplace transform of the function $F_1(t) = Q_R \; 0 \; X_R$ is the product $f(p)g(p)$ and hence the transform of the series, Equation [10], is

$$\varphi(p) = \frac{1}{1 + f(p)g(p)} = \frac{F(p)}{G(p)} \qquad [13]$$

In order to find the behavior function X_B, of which Equation [13] is the Laplace transform, use may be made of Heaviside's "expansion theorem," which is stated as follows:

If $p_1, p_2, \ldots, p_n$ are the roots of the equation $G(p) = 0$, each assumed to be different and none of them equal to zero, then Equation [13] is the Laplace transform of

$$X_B(t) = \frac{F(0)}{G(0)} + \Sigma \frac{F(p_j)}{p_j G'(p_j)} e^{p_j t}$$

The expansion theorem can be applied when $F(p)$ and $G(p)$ are polynomials, and also in the case of transcendental functions, where it becomes impracticable, however, for small values of t.

In the applications here considered, the Laplace transform of $Q_R = a + bt$ is

$$g(p) = a + \frac{b}{p} \qquad [14]$$

As to $X_R(t)$, we will first consider the cases in which its Laplace transform $f(p)$ is a rational function (Types A and D). Next we shall deal with the cases in which $f(p)$ is a transcendental function (Types B, C, and E).

Formulas for S. When $f(p)$ is a rational function, the Laplace transform of $X_B(t)$ is a rational function

$$\varphi(p) = \frac{p(d_0 p^{n-1} + d_1 p^{n-2} + \ldots + d_{n-1})}{p^n + c_1 p^{n-1} + \ldots + c_n}$$

It is possible to find a general expression for the integral

$$S = \int_0^\infty X_B{}^2(t)dt$$

for any function $X_B(t)$ whose Laplace transform is a rational fraction

$$\varphi(p) = \frac{pE(p)}{(p - \alpha_1)(p - \alpha_2) \ldots (p - \alpha_n)}$$

provided all the roots α_i of the denominator have negative real parts. The degree of $E(p)$ must not be higher than $n - 1$. The factor p in the numerator is necessary in order that $X_B(t)$ tends to zero for $t \to \infty$.

According to the expansion theorem, we have

$$X_B(t) = \sum_j \frac{E(\alpha_j) \exp(\alpha_j t)}{(\alpha_j - \alpha_1) \ldots (\alpha_j - \alpha_{j-1})(\alpha_j - \alpha_{j+1}) \ldots (\alpha_j - \alpha_n)}$$

and the value of the integral S is expressed by

$$S = -\sum_i \frac{E_2(\alpha_j)}{2\alpha_j \prod\limits_{k \neq j} (\alpha_j - \alpha_k)^2}$$

$$-2 \sum_{(ij)} \frac{E(\alpha_i)E(\alpha_j)}{(\alpha_i + \alpha_j) \prod\limits_{k \neq i} (\alpha_i - \alpha_k) \prod\limits_{k \neq j} (\alpha_j - \alpha_k)}$$

It may be proved that S is a symmetrical function of the roots α_i, and thus can be expressed as a rational function of the coefficients c_k. Actually, S may be found more easily by means of a theorem to be stated later (Equation [15]). The general expression for S, which is rather long and complicated, is not given here but is only written down for those values of n which are of practical interest. Also we have given the conditions of stability, according to the Hurwitz criterion

$n = 2$

$$S = \frac{d_1{}^2 + c_2 d_0{}^2}{2c_1 c_2}$$

$n = 3$

$$S = \frac{d_2{}^2 c_1 + (d_1{}^2 - 2d_0 d_2)c_3 + c_2 c_3 d_0{}^2}{2c_3(c_1 c_2 - c_3)}$$

Condition of stability

$$c_1 c_2 - c_3 > 0$$

$n = 4$

$$S = \frac{d_3{}^2(c_1 c_2 - c_3)}{2c_4(c_1 c_2 c_3 - c_3{}^2 - c_1{}^2 c_4)}$$

$$+ \frac{c_1(d_2{}^2 - 2d_1 d_3) + c_3(d_1{}^2 - 2d_0 d_2) + d_0{}^2(c_2 c_3 - c_1 c_4)}{2(c_1 c_2 c_3 - c_3{}^2 - c_1{}^2 c_4)}$$

Condition of stability

$$c_1 c_2 c_3 - c_3{}^2 - c_1{}^2 c_4 > 0$$

$n = 5$

$$S = \frac{d_4{}^2 \omega_0}{2c_5 G_5} +$$

$$\frac{(d_3{}^2 - 2d_2 d_4)\omega_1 + (d_2{}^2 - 2d_1 d_3 + 2d_0 d_4)\omega_2 + (d_1{}^2 - 2d_0 d_2)\,\omega_3 + d_0{}^2 \omega_4}{2G_5}$$

$$\omega_0 = c_1 c_2 c_4 - c_3{}^2 - c_1{}^2 c_4 + c_1 c_5$$

$$\omega_1 = c_1 c_2 - c_3$$

$$\omega_2 = c_1 c_4 - c_5$$

$$\omega_3 = c_3 c_4 - c_2 c_5$$

$$\omega_4 = c_2 c_3 c_4 - c_2{}^2 c_5 - c_1 c_4{}^2 + c_4 c_5$$

$$G_5 = \omega_1 \omega_3 - \omega_2{}^2$$

Condition of stability

$$G_5 > 0$$

$n = 6$

$$S = \frac{d_5{}^2 \omega_0}{2c_6 G_6} + \frac{(d_4{}^2 - 2d_3 d_5)\omega_1 + (d_3{}^2 - 2d_2 d_4 + 2d_1 d_5)\omega_2}{2G_6}$$

$$+ \frac{(d_2{}^2 - 2d_1 d_3 + 2d_0 d_4)\omega_3 + (d_1{}^2 - 2d_0 d_2)\omega_4 + d_0{}^2 \omega_5}{2G_6}$$

$$\omega_0 = (c_3 c_4 - c_2 c_5 + c_1 c_6)\,(c_1 c_2 - c_3) - (c_1 c_4 - c_5)^2$$

$$\omega_1 = c_1 c_2 c_3 - c_3{}^2 - c_1{}^2 c_4 + c_1 c_5$$

$$\omega_2 = c_1 c_2 c_5 - c_3 c_5 - c_1{}^2 c_6$$

$$\omega_3 = c_1 c_4 c_5 - c_1 c_3 c_6 - c_5{}^2$$

$$\omega_4 = c_3 c_4 c_5 - c_3{}^2 c_6 - c_2 c_5{}^2 + c_1 c_5 c_6$$

$$\omega_5 = (c_2 c_3 - c_1 c_4 + c_5)\,(c_4 c_5 - c_3 c_6) - (c_1 c_6 - c_2 c_5)^2$$

$$G_6 = c_5 \omega_0 - c_3 c_6 \omega_1 + c_1 c_6 \omega_2$$

Conditions of stability

$$\omega_1 > 0$$

$$G_6 > 0$$

For higher values of n the expressions become too cumbersome for practical use. In some of these cases S may be found from the integral

$$S = \frac{1}{\pi}\int_0^\infty |\varphi(i\omega)|^2 \frac{d\omega}{\omega^2} \qquad [15]$$

which follows from Parseval's theorem in the theory of the Fourier integral.

Application of Formulas for S to Types of Process A and D

Type A. The response of the process is given by Equation [3]. The Laplace transform is

$$f(p) = \frac{1}{(p+1)^n}$$

hence by Equations [13] and [14]

$$\varphi(p) = \frac{p(p+1)^n}{p(p+1)^n + ap + b} \qquad [16]$$

Applying the formulas for S to the present case, Equation [16], we find, putting $a + 1 = k$

for $n = 1$

$$S = \frac{b+1}{2bk}$$

$n = 2$

$$S = \frac{bk + 2b + 2}{2b(2k - b)}$$

$n = 3$

$$S = \frac{9 - k + 9b + 6bk - 3b^2}{2b(9k - k^2 - 9b)}$$

$n = 4$

$$S = \frac{80 - 16k + 84b + 64bk - 66b^2 - 4bk^2 + b^2k}{2b[80k - 120b - (4k - b)^2]}$$

$n = 5$

$S =$

$$\frac{-(155b+1)x^2 + (23b^2 + 260b - 70)x - b^3 - 8b^2 + 63b + 55}{10b[-x^3 - 70x^2 + (14b + 55)\,x - b^2 - 22b]}$$

In the last case ($n = 5$), we have put $k = 5x$.

The values of a or k and b for which S becomes a minimum may be found by solving the equations

$$\frac{\partial S}{\partial k} = 0, \qquad \frac{\partial S}{\partial b} = 0$$

It appears that there is only one set of roots of these equations which satisfies the conditions of stability.

For a table of values of a and b, we refer to the companion paper[2] (Table 1).

For larger values of n, the minimum of S can be found approximately by means of the integral, Equation [15].

For the case $n = 4$ (third-order contact), the lines of constant N in the a, b-plane are shown in Fig. 3. It is seen that deviations of 40 per cent in a and b do not bring the quality number below 0.9. Thus a very rigorous adjustment of a and b is not necessary.

Type D (Without Self-Regulation). The method is exactly the same as in the preceding case. The Laplace transforms of the functions X_R and X^*_B are

$$f(p) = \frac{1}{p(p+1)^n}$$

$$\varphi(p) = \frac{p(p+1)^n}{p^2(p+1)^n + ap + b}$$

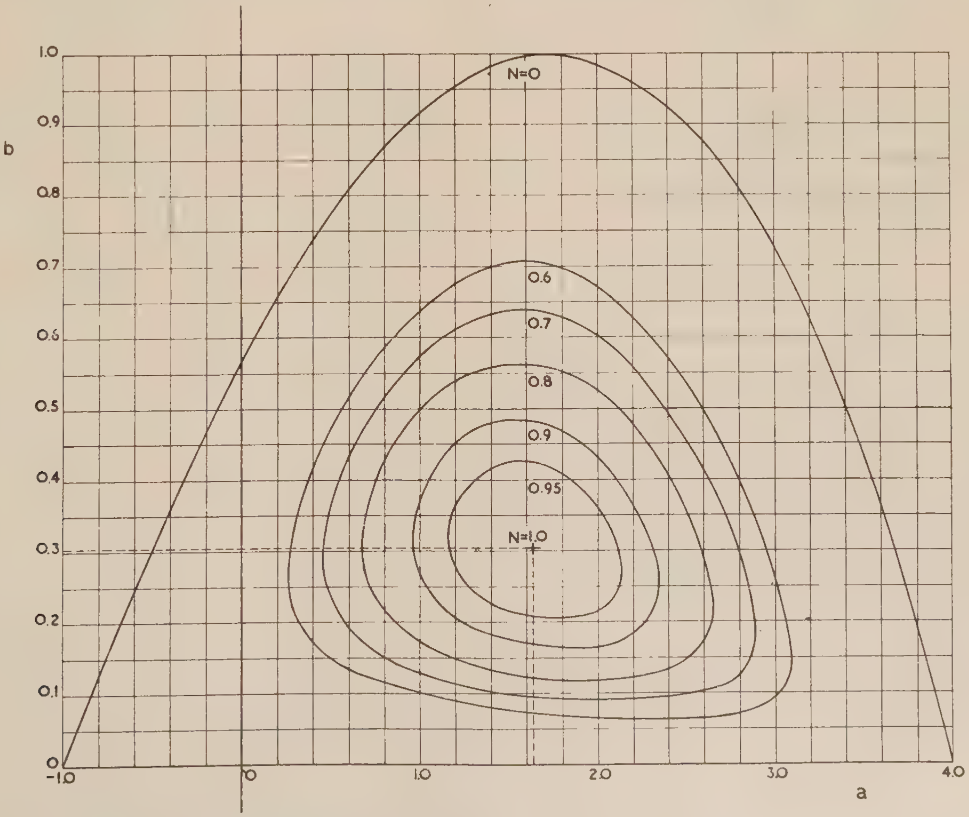


Fig. 3 Lines of Equal Quality of Regulation for Case $n = 4$ of Type A Process

$\left(N = \frac{S_m}{S}\right)$

$n = 0$

$$S = \frac{1}{2ab}$$

$n = 1$

$$S = \frac{1 + b}{2b(a - b)}$$

$n = 2$

$$S = \frac{2 - a + 4b + ab}{2b(2a - a^2 - 4b)}$$

$n = 3$

$$S = \frac{8 - 9a + 27b + 10ab - 6b^2}{2b(8a - 24b - 9a^2 + 6ab - b^2}$$

$n = 4$

$$S = \frac{64 - 112a + 336b - a^2 + 156ab - 176b^2 - 10a^2 b + 4ab^2}{2b(64a - 256b - 112a^2 + 144ab - 64b^2 - a^3)}$$

The values of a and b for which these expressions assume a minimum are given in Table 3 of the companion paper.[2]

PROCESSES OF TYPES B, C, AND E

In these cases the operational expression $\varphi(p)$ for $X_B(t)$ is transcendental and the determination of the most favorable values of a and b is far more troublesome, as the minimum value of S can only be found approximately by trial and error. For this purpose we have made use of the integral, Equation [15].

Type B. The response of the process is given by Equation [4]. Assuming $T_e = 1$, the Laplace transform of X_R is

$$f(p) = \frac{\exp(-pT_d)}{1 + p}$$

hence

$$\varphi(p) = \frac{p(p + 1)}{p(p + 1) + (ap + b)\exp(-pT_d)}$$

In this case the integral, Equation [15], becomes

$$S = \frac{1}{\pi}\int^{\infty} \frac{(1 + \omega^2)d\omega}{\omega^2(a + \cos\omega T_d - \omega\sin\omega T_d)^2 + (b - \omega\sin\omega T_d - \omega^2\cos\omega T_d)^2}$$

In order to determine the minimum value of S, a and b have been systematically varied, and the integral has been evaluated numerically. It is only necessary to consider a finite interval from 0 to ω_0 where ω_0 must be assigned a suitable value depending upon the value of T_d. The contribution to S by the interval from ω_0 to ∞ remains nearly constant when a and b are varied, and thus does not materially affect the optimum values for a and b. It is again found that the quality of regulation remains good for a considerable latitude of variation of a and b.

The results of the calculation are recorded in Table 2 of the companion paper.[2]

After this somewhat more detailed exposition, we may be short about the remaining Types C and E. The best values of a and b are given in the companion paper.[2]

STABILITY

We conclude with a few remarks on stability of processes with dead time T_d.

When the process response has a dead time T_d the Laplace transform may be written

$$f(p) = e^{-pT_d}F(p)$$

and the Laplace transform of the behavior function $X_B(t)$ is

$$\varphi(p) = \frac{1}{1 + e^{-pT_d}F(p)\left(a + \frac{b}{p}\right)}$$

$$= \frac{p}{G(p)}$$

The denominator of this function has an infinite number of roots, all of which should have negative real parts in order that regulation may be stable. Regulation becomes unstable for values of a and b for which a root crosses the imaginary axis. Moreover, b should be positive, as for negative values of b there would be a positive root.

In order to find the condition that $G(p)$ has purely imaginary roots, we substitute $p = i\omega$ and put $G(i\omega) = G_1(\omega) + iG_2(\omega) = 0$. Thus, a and b are expressed as functions of the parameter ω. These equations determine a spiral-shaped curve $\psi(a, b) = 0$ in the (a, b)-plane. The area enclosed by the smallest loop of this curve and the axis $b = 0$ is the region of stability.

As an example we consider a process of Type E with $T_d = 1$. The Laplace transform of $X^*_B(t)$ is in this case

$$\varphi(p) = \frac{p}{p^2 + e^{-p}(ap + b)}$$

and the limits of stability are given by the line $b = 0$, and the smallest loop of the curve

$$\begin{cases} a = \omega\sin\omega \\ b = \omega^2\cos\omega \end{cases}$$

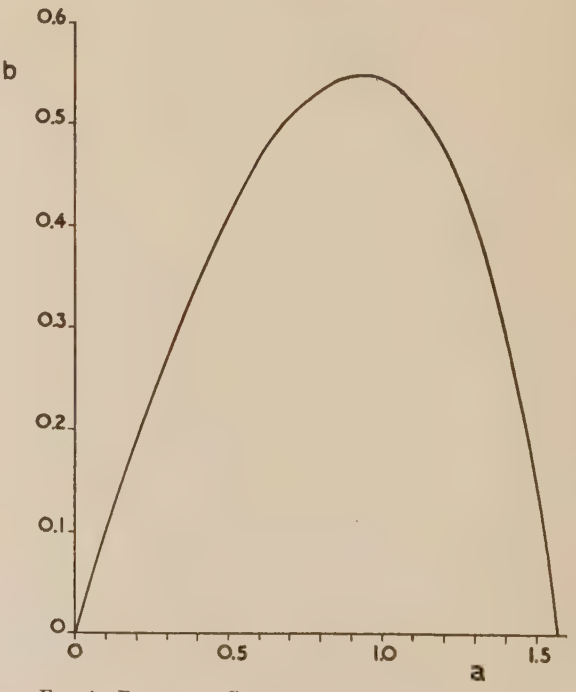


FIG. 4 REGION OF STABILITY FOR TYPE E PROCESS

ACKNOWLEDGMENT

Our acknowledgments are due to the N. V. de Bataafsche Petroleum Maatschappij for permission to publish this paper.

Discussion

M. D. Creech.[4] There are two requirements to be met in obtaining satisfactory control: One is that the amplitude of the controlled variable remain small; the other is that the amplitude approach zero at an early time. The integral of the absolute value of the amplitude gives about equal weight to the amplitude and the rapidity with which the amplitude approaches zero. As was pointed out by the authors, the integral of the square of the amplitude is more suitable to mathematical treatment. However, the integral of the square of the amplitude gives more weight to the magnitude of the amplitude and less weight to the time it takes for oscillations to die out. It now occurs to the writer that perhaps it might be better to use the integral of the fourth power or sixth power of the amplitude in certain cases.

An answer is requested to the following question: How can one tell when optimum control is obtained? Is it not possible that, in one case where the amplitude is large but quickly approaches zero and in another where the amplitude is small but takes a long time to become small, the minimum integral could be the same in both cases?

AUTHORS' CLOSURE

It may theoretically be admitted there might be some reason for the use of other than second powers in our minimum integral criterion, depending upon the relative weight of duration and amplitude of the regulated disturbance. It should however be recalled that: first, owing to several circumstances, our theory is only an approximation to any actual regulation; second, the optimum values of regulator parameters are not very critical, approximately forty per cent deviation from these values scarcely reducing the quality of regulation. For these reasons, apart from mathematical difficulties, it seems an unnecessary refinement to consider other than second powers of amplitude, the more so as it would be difficult to decide which power should be chosen in any particular case.

The question whether our integral might have two approximately equal minimum values may be answered by the fact that in the cases we investigated, the function represented by our integral had only one single minimum, to which corresponds a definite set of values of regulation parameters. Hence in all these cases the behavior of the regulation circuit is uniquely determined. As our theory is only to be applied to regulation of processes whose response curves are sufficiently closely approximated by our hypothetical response curves, it may be expected also for these processes that optimum regulation according to our criterion is obtained for a unique set of values of regulation parameters. It may be granted that the case of two or even more minima of our integral is not a priori impossible, for example when there is more than one region of stability; it is, however, doubtful if such a case would ever be met with in practice. Anyway such a process would be beyond the scope of our theory.

At this place, reference may be made to "The Theory of Servo-Mechanisms," by James, Nichols, and Phillips, 1947, (McGraw-Hill Book Co., 1947), which came to the authors' attention after their papers had been submitted for publication. In a section of this book an approach to the control problems in many ways similar to that of the authors has been followed. The authors would like to avail themselves of this opportunity to express their thanks to Mr. N. B. Nichols of Taylor Instrument Co., firstly, for the stimulating influence of his repeatedly cited paper on "Optimum Settings for Automatic Controllers"; secondly, for discussions with members of our staff, which have resulted in considerable changes in our original design, especially of the companion paper; and thirdly, for his kindness in presenting these papers.

---

[4] Professor, Mechanics and Metallurgy, The University of Oklahoma, Norman, Okla. Mem. ASME.

The Optimum Adjustment of Regulators

By P. HAZEBROEK[1] and B. L. van der WAERDEN[1]

In this paper a practical method is proposed for the adjustment of regulators in order to obtain the most satisfactory regulation of a process variable which should be kept at a constant value. The regulator is assumed to perform proportional and integral action, derivative action not being taken into consideration for the present.

INTRODUCTION

So far, little attention has been paid to the problem of assigning definite values to the parameters characterizing the action of a regulator in order to obtain optimum results. However, an important paper by J. H. Ziegler and N. B. Nichols,[2] has been published, which gives some simple rules and which later on will be considered more closely.

The present authors, starting from other fundamentals, have developed new conditions for the adjustment of parameters. The novel feature of their theory is the proposal of a "general criterion for best regulation." This paper, which is intended to bring their theory within the scope of practical application, can be understood independently of the underlying theoretical foundation, which will be given in a separate paper.[3]

FUNDAMENTAL NOTIONS

Before considering the subject proper the authors will recall some well-known fundamental notions to be used later.

Let x be a "regulated variable," e.g., a temperature, pressure, etc., which should be kept as constant as possible. The value of this variable is obtained by measurement. The deviation from the regulation point, i.e., from the normal value, may be identified with the displacement of the recording pen from this point, measured in arbitrary units. The regulation point will be taken as zero point and thus x is the deviation from zero.

Let q be the "regulating flow." It can be measured by the position of the final regulating element, e.g., a regulating valve, or by the air pressure regulating it, assuming a linear relationship between them. Again, the normal value assumed by q in the absence of any disturbance is taken to be $q = 0$. The sign of q is made such that positive deviations of q cause positive deviations of x.

Process and regulator constitute a closed system which will be called a "regulation circuit", Fig. 1.

The "response of the process" is defined as follows: Let the valve be given at time $t = 0$ a sudden displacement q_0 and let it be kept in this position, the valve being disconnected from the regulator. The valve movement q_0 will cause in course of time a change in the regulated variable x, which is assumed to be proportional to the valve movement q_0. The ratio x/q_0 as a function of time is called the "process response function" and is denoted by $X_R(t)$. Thus, in other words, we may say that $X_R(t)$ is the deviation of x due to a sudden disturbance $q_0 = 1$. This latter disturbance will be called "unit" or "standard disturbance."

It follows from this definition that the process-response curve results from the properties both of the process and of the measuring part of the regulator.

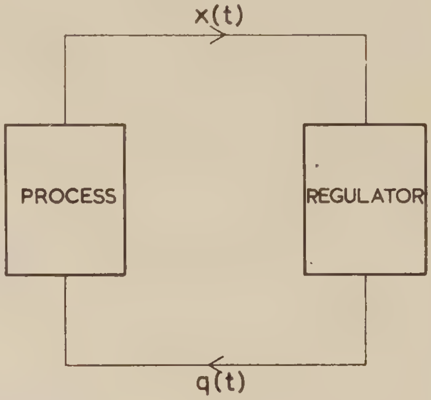


FIG. 1 DIAGRAM OF REGULATION CIRCUIT

The "response of the regulator" is defined as follows: Let the pen be given at time $t = 0$ a sudden displacement x_0 from its normal position and let the pen be kept in this position, the process being put out of circuit. The regulator will interpret the movement of the pen into a need for corrective action that is, the regulating flow $q(t)$ will become negative in order to compensate a positive x_0. This change is assumed to be proportional to x_0; its opposite $-q(t)$ divided by x_0 is called the "response function of the regulator" $Q_R(t)$.

The response of a regulator with proportional plus integral action is represented by a function of the form

$$Q_R(t) = a + bt$$

From the response functions of process and regulator we may derive two functions describing the effect of regulation in the "closed" circuit. A sudden displacement x_0 of the recording pen will result in a response $x(t)$ of the regulated variable, and a response $q(t)$ of the regulating flow. The quotient $x(t)/x_0 = X_B(t)$ will be called the "behavior of the regulated variable," the quotient $-q(t)/x_0 = Q_B(t)$, the "behavior of the regulating flow."

It may happen that the theoretically best values of the regulation parameters would give rise to too violent fluctuations of the regulating flow. In this case Q_B must satisfy certain restricting conditions in consequence of which the said parameters are not allowed to exceed certain limiting values. These conditions depend upon the nature of the particular plant under consideration. As it is difficult to give satisfactory general rules, this point is merely mentioned here but henceforth will be left out of account.

CLASSIFICATION OF PROCESSES

By considering the process-response functions we can distinguish two classes of processes:

1 Those processes in which the regulated variable tends to a

---

[1] N. V. de Bataafsche Petroleum Maatschappij, The Hague, Holland.

[2] "Optimum Settings for Automatic Controllers," by J. H. Ziegler and N. B. Nichols, Trans. ASME, vol. 64, 1942, p. 759.

[3] "Theoretical Considerations on the Optimum Adjustment of Regulators," by P. Hazebroek and B. L. van der Waerden, companion paper being presented at the present IIRD-ISA Meeting.

Presented at the ASME Industrial Instrument and Regulators Division–Instrument Society of America Conference, St. Louis, Mo., September 12–16, 1949.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 49—IIRD-2.

new fixed value, when a standard disturbance is applied to the regulating flow (processes "with" self-regulation).

2 Those processes in which the regulated variable increases or decreases indefinitely with time under the influence of a standard disturbance in the regulating flow (processes "without" self-regulation). We will consider especially the important case in which the increase is linear after a certain lapse of time.

TYPES OF DISTURBANCES

When defining the behavior functions as mentioned, a standard disturbance x_0 is assumed to be given to the regulated variable. In practice, however, the disturbances occurring in a process usually do not originate in the measuring part of the circuit, but may be introduced at any point in the process part. Hence the disturbance in the regulated variable in most cases is not a standard disturbance, but responds to some transformation of the initial disturbance somewhere else in the process.

We will distinguish the disturbances occurring in the regulated variable according to their asymptotic behavior for large values of t:

1 The disturbance tends to zero after a certain lapse of time.

2 The disturbance tends asymptotically to a fixed value. The simplest disturbance of this kind is the standard disturbance, introduced previously.

3 The disturbance increases indefinitely with time.

POSSIBLE COMBINATION OF PROCESSES AND DISTURBANCES

Each of the disturbances just classified may occur in each of the two classes of processes previously mentioned.

As the type of regulator to be chosen and its adjustment for optimum regulation depend on which of these cases we are considering, all these cases should be discussed from a theoretical point of view. In practice, however, two of the theoretically possible combinations are especially important. We will confine ourselves in this paper to these cases, which are as follows:

1 Processes with self-regulation, when a disturbance of the second kind is applied to the regulated variable. As will be shown later, it is sufficient for our purpose to consider the standard disturbance.

2 Processes without self-regulation whose response functions increase linearly with time, when a disturbance $x = t$ is applied to the regulated variable (special case of third kind of disturbance). An example is the regulation of a liquid level. The disturbance then usually results from a sudden change of the regulating flow, which is transformed into a linearly increasing disturbance of the regulated variable.

In this second case the behavior function $X_B(t)$, corresponding to a standard disturbance in the regulated variable, has mainly a theoretical significance. The response to a disturbance

$$x(t) = \begin{cases} 0 \text{ for } t < 0 \\ t \text{ for } t > 0 \end{cases}$$

(time integral of a standard disturbance) is far more important for judgment of the behavior of the regulated variable. This response function, which we shall call the "second behavior function" $X_B{}^*(t)$, is the time integral of $X_B(t)$

$$X_B{}^*(t) = \int_0^t X_B(\tau)d\tau$$

CONSTANTS OF REGULATOR

Two constants characterize the response function of the type of regulator we are discussing: the sensitivity[4] of proportional response and the "reset rate" b/a.

Sensitivity is concerned only with the first (vertical) part of the response curve; it is valve movement per pen movement. In practice it is customary to use in place of sensitivity the proportional band, defined as

$$PB = \frac{1}{a} \cdot \frac{q_m}{x_m} \cdot 100$$

where

q_m = maximum interval of the regulating flow
x_m = scale interval of the regulated variable

The reset rate is concerned with the second part of the curve; it is the number of times per unit of time that automatic reset repeats the proportional response a. Hence the reset rate b/a has the dimension of an inverse time; in fact, the reciprocal a/b, the reset time T, is the time in which proportional response is repeated, see Fig. 2.

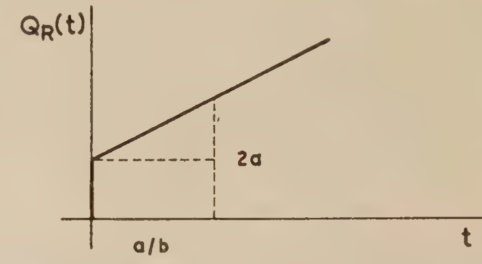


FIG. 2 RESPONSE CURVE OF REGULATOR OF PROPORTIONAL-PLUS-INTEGRATING TYPE

CRITERION FOR BEST REGULATION

Regulators with proportional plus integral action allow an adjustment of the parameters a and b/a within relatively wide limits. Hence the question arises: Which values of a and b/a will give the best regulation, reducing any disturbance of x to zero within a short time, without giving rise to violent fluctuations?

Before it is possible to answer this question, it should first be decided what we are to consider as the best possible regulation. The authors have adopted the following criterion, the justification of which they have given in their theoretical paper:[3]

1 For processes with self-regulation, optimum regulation will be obtained for those values of the parameters a and b for which the time integral of the square of the behavior of the regulated variable

$$S = \int_0^\infty X_B{}^2\, dt$$

attains its minimum value.

2 For processes without self-regulation, the time integral of the square of the second behavior function

$$S = \int_0^\infty X_B{}^{*2}\, dt$$

should be minimized.

We have applied this criterion to a number of hypothetical process-response curves representing mathematical functions with particularly simple properties. For the details of the calculations we refer to our theoretical paper.[3] These curves, together with the values of a and b, corresponding to optimum regulation, are given in Table 1.

---

[4] Sometimes sensitivity is identified with proportional band, which seems to us a confusing practice.

TABLE 1

n	a	b
2	∞	∞
3	2.75	0.500
4	1.64	0.305
5	1.26	0.219
10	0.692	0.0913
25	0.555	0.0330
50	0.516	0.0162

Processes With Self-Regulation:

Type A

$$X_R(t) = 1 - e^{-t}\left(1 + t + \frac{t^2}{2!} + \ldots + \frac{t^{n-1}}{(n-1)!}\right)$$

(Fig. 3 for $n = 4$)

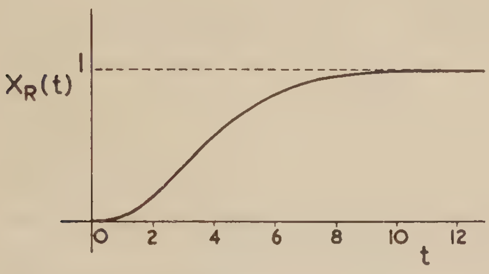


FIG. 3 RESPONSE CURVE OF TYPE A PROCESS

Type B

$$X_R(t) = \begin{cases} 0 & \text{for } t < T_d \\ 1 - \exp\left(-\dfrac{t - T_d}{T_e}\right) & \text{for } t > T_d \end{cases} \quad \text{(Fig. 4)}$$

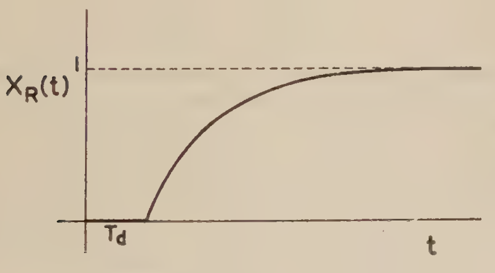


FIG. 4 RESPONSE CURVE OF TYPE B PROCESS

For time lag $T_d = 1$, we find approximately for different values of the time constant, data in Table 2.

TABLE 2

$\frac{1}{T_e}$	a	b
0.2	4.0	0.53
0.3	2.77	0.54
0.5	1.79	0.55
0.7	1.37	0.56
1	1.07	0.57
1.5	0.84	0.60
2	0.73	0.62
3	0.63	0.66
5	0.55	0.68
∞	0.50	0.80

Type C. Abrupt response with dead time T_d

$$X_R(t) = \begin{cases} 0 \text{ for } t < T_d \\ 1 \text{ for } t > T_d \end{cases} \quad \text{(Fig. 5)}$$

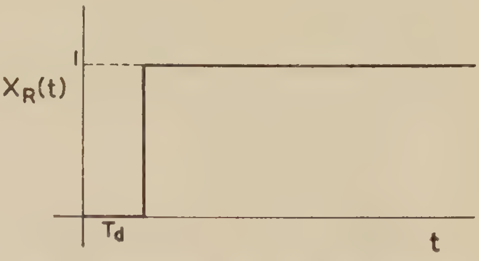


FIG. 5 RESPONSE CURVE OF TYPE C PROCESS

This is a limiting case of Type A when n becomes infinite and the time scale is transformed into $t' = T_d\, t/n$. It is also a limiting case of Type B when T_e becomes infinite. The most favorable values of a and b are

$$a = 0.50, \quad b = \frac{0.8}{T_d}$$

Processes Without Self-Regulation:

Type D

$$X_R(t) = t - n + e^{-t}\left[n + (n-1)t + \frac{n-2}{n!}t^2 + \ldots + \frac{1}{(n-1)!}t^{n-1}\right]$$

(Fig. 6 for $n = 5$)

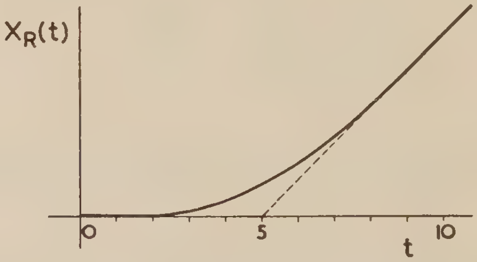


FIG. 6 RESPONSE CURVE OF TYPE D PROCESS

Best values of a and b are given in Table 3.

TABLE 3

n	a	b
0 and 1	∞	∞
2	1.211	0.0917
3	0.571	0.0335
4	0.375	0.0174

Type E

$$X_R(t) = \begin{cases} 0 & \text{for } t < T_d \\ t - T_d & \text{for } t > T_d \end{cases} \quad \text{(Fig. 7)}$$

Best values of a and b

$$a = \frac{1.11}{T_d}, \quad b = \frac{0.24}{T_d^2}$$

Type E may be obtained as a limiting case of Type D for $n \to \infty$ after having reduced both the time scale and the scale of the regulated variable by a factor T_d/n.

CHARACTERISTIC PARAMETERS OF PROCESS RESPONSE CURVES INTRODUCED BY ZIEGLER AND NICHOLS

In order to apply these theoretical results to the response curves produced by actual processes, it is necessary to define

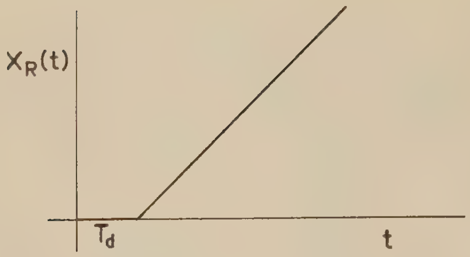


Fig. 7 Response Curve of Type E Process

certain quantities which determine to a more or less close approximation the shape of the response curve. In this we follow the same lines as Ziegler and Nichols.

Ziegler and Nichols introduce two parameters occurring in a process-response curve (see Fig. 8)

R_1 = maximum slope of curve = time derivative in point of inflection[5]

L = time lag: distance along the axis from origin to point of intersection with inflection tangent.

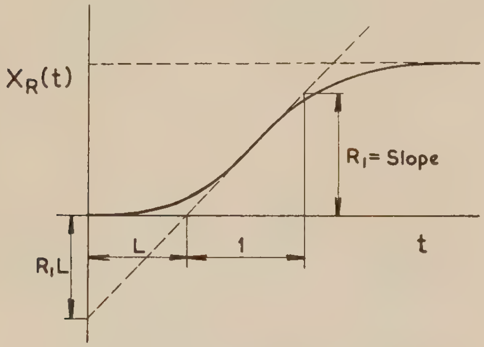


Fig. 8 Characteristic Parameters of Process-Response Curve According to Ziegler and Nichols

The parameters a and b/a of the regulator are now to be expressed in the introduced process parameters, in order to adapt the response curve of the regulator to the process-response curve.

Whereas L has the dimension of a time, the dimension of R_1L is "pen movement per valve movement." The inverse values $1/L$ and $1/(R_1L)$ have just the dimensions required for the reset rate and the sensitivity of a regulator. Hence Ziegler and Nichols define the optimum values of a and b/a by the formulas

$$\text{Sensitivity } a = \frac{0.9}{R_1L} \qquad [1]$$

$$\text{Reset rate } \frac{b}{a} = \frac{0.3}{L} \qquad [2]$$

The factors 0.9 and 0.3 are based on data obtained from practical experience.

Comparison of Ziegler and Nichols' Results With Present Authors' Results

We have applied Ziegler and Nichols' rules to a set of our response curves Type B. Each of the curves is characterized by its "time constant" T_c.

Taking T_d as time unit, the values to be substituted in the formulas of Ziegler and Nichols are

$$\left.\begin{aligned} L &= T_d = 1 \\ LR_1 &= T_d \cdot \frac{1}{T_c} = \frac{1}{T_c} \end{aligned}\right\} \qquad [3]$$

Hence the optimum sensitivity and reset rate would be, according to those authors

$$a = \frac{0.9}{LR_1} = 0.9\, T_c \qquad [4]$$

$$\frac{b}{a} = \frac{0.3}{L} = 0.3 \qquad [5]$$

For comparison, the authors' results and those of Ziegler and Nichols are recorded side by side in Table 4.

TABLE 4 COMPARISON OF AUTHORS' RESULTS AND THOSE OF ZIEGLER AND NICHOLS

	Authors' results		Ziegler and Nichols' results		
$\frac{1}{T_c}$	a	$\frac{b}{a}$	a	$\frac{b}{a}$	
0.2	4.0	0.13	4.5	0.3	Reset rate high
0.3	2.77	0.195	3.00	0.3	Good agreement
0.5	1.79	0.31	1.80	0.3	
0.7	1.37	0.41	1.28	0.3	
1.0	1.07	0.53	0.90	0.3	Reset rate low
1.5	0.84	0.71	0.60	0.3	
2.0	0.73	0.85	0.45	0.3	Reset rate and sensitivity low
3.0	0.63	1.05	0.30	0.3	
5.0	0.55	1.24	0.18	0.3	
∞	0.50	1.38	0	0.3	

We have found that in all the cases we have computed, the minimum for S is very flat, so that for considerable deviations of a and b from their optimum values regulation still remains satisfactory. In our theoretical paper,[3] we have given an instance where, even with deviations of 40 per cent, regulation remains good and, in other cases we have investigated, the conditions were found to be quite similar. Taking this into account, it may be stated that for $T_d < T_c$ (except when T_d/T_c is very small) the values of a and b found by both methods agree very well. For larger T_d/T_c, however, the reset rate given by Ziegler and Nichols' rule is far too low, and so is the sensitivity a for still larger T_d/T_c.

This conclusion is in accordance with a statement of Ziegler and Nichols in a later paper:[6] "These settings appear to be very nearly correct on the processes tested for wide variations of R and L. On those infrequent processes in which L becomes greater than Z, Fig. 2,[7] the settings given are too conservative."

Now L is our T_d and Z is our T_c, hence the "infrequent processes" to which Ziegler and Nichols' rules are admittedly not applicable are just the cases with $T_d/T_c > 1$. The cases with small lags ($T_d < T_c$) are those in which regulation is easy, whereas great lags make it difficult. As we shall see, it is possible to modify the rules of Ziegler and Nichols in such a way that they become generally applicable also in cases where the time lag is large.

Modification and Extension of Concepts Introduced by Ziegler and Nichols

Ziegler and Nichols' formulas cannot be improved as long as only the constants L and R_1, lag and slope of the process-response curve, are taken into account. Hence in order to give rules with a wider range of application, a third constant must be introduced. We can choose for this constant, in the case of processes with self-regulation in which x tends to a finite ultimate value, either the *"height"* H, of the horizontal asymptote or the "second lag" L_2, described by Ziegler and Nichols as the time difference between the points of intersection of the inflection tangent with the t-axis and the asymptote (Fig. 9). H and L_2 are connected by the relation

---

[5] Ziegler and Nichols first introduce a quantity $R = \max \frac{dx}{dt}$ for arbitrary q_0, but afterward they divide it by q_0, and thus obtain the "unit reaction rate" R_1.

[6] "Process Lags in Automatic-Control Circuits," by J. G. Ziegler and N. P. Nichols, Trans. ASME, vol. 65, 1943, p. 435.

[7] Compare Fig. 9 of this paper where $L_2 = Z$.

$$H = R_1 L_2$$

For processes without self-regulation, H and L_2 are both ∞.

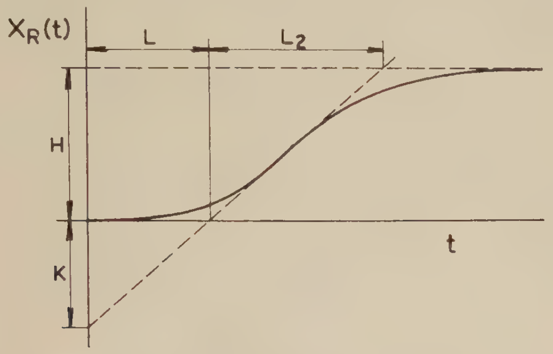


FIG. 9 EXTENDED DEFINITION OF CHARACTERISTIC PARAMETERS OF PROCESS-RESPONSE CURVE

Owing to the difference in definition of the criterion for best regulation, processes with and without self-regulation must be treated separately.

PROCESSES WITH SELF-REGULATION

The second lag L_2 has, like L, the dimension of a time. Hence the lag quotient

$$\lambda = \frac{L}{L_2} = \frac{R_1 L}{H} = \frac{K}{H} \quad \ldots\ldots\ldots\ldots\ldots\ldots [6]$$

is a dimensionless quantity. The product R_1L has a simple geometrical meaning; it is the "distance" K, measured from the origin down to the inflection tangent.

We now replace Equations [1] and [2] by the more general ones

$$\text{Sensitivity } a = \frac{\alpha}{R_1 L} = \frac{\alpha}{K} \quad \ldots\ldots\ldots\ldots\ldots [7]$$

$$\text{Reset rate } \frac{b}{a} = \frac{\beta}{L} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [8]$$

where the dimensionless quantities α and β are functions of λ, and are assigned such values that for curves Type B the Equations [7], [8] yield the exact optimum values of Table 2. For these curves the lag quotient is

$$\lambda = \frac{T_d}{T_e}$$

and since a, b, R_1, and L are known in each of the ten cases, the coefficients α and β can be calculated from Equations [7] and [8], with the results given in Table 5.

TABLE 5 OPTIMUM VALUES OF a AND b FOR TYPE B RESPONSE

λ	α	β
0.2	0.80	0.14
0.3	0.83	0.20
0.5	0.89	0.31
0.7	0.96	0.41
1.0	1.07	0.54
1.5	1.26	0.71
2.0	1.46	0.85
3.0	1.89	1.05
5.0	2.75	1.23
∞	∞	1.60

For $\lambda < 0.2$

$$\alpha = 0.74 + 0.3\,\lambda, \quad \beta = 0.7\,\lambda$$

The case $\lambda = 0$ requires some comments. It will be seen from Equation [6] that λ may become zero in different ways, as follows:

1 When L becomes zero, which corresponds to a process with self-regulation and with dead time = 0.

2 When H becomes infinite, corresponding to a process without self-regulation and having a dead time different from zero.

Only the first possibility is of interest. We have seen that for processes without self-regulation our criterion for best regulation should be modified, so that the values for $\lambda = 0$, $\alpha = 0.74$, $\beta = 0$ are of no use for this second case.

TEST OF IMPROVED FORMULAS

In order to test whether Equations [7], [8] are also applicable to other process-response curves than those of Type B, for which they have been derived, we have applied them to Type A curves without dead time.

The optimum values of a and b have been given in Table 1 and the corresponding values of λ, α, and β, which can be found analytically, are recorded in Table 6.

TABLE 6 OPTIMUM VALUES FOR TYPE A RESPONSE

n	λ	α	β
3	0.218	0.600	0.146
4	0.319	0.523	0.265
5	0.410	0.526	0.366
10	0.773	0.535	0.774
25	1.500	0.832	1.10
50	2.326	1.200	1.28

For comparison, the values of α and β for both Type A and Type B response are represented graphically in Figs. 10 and 11. For small values of λ, comparison is impossible, because in the case of the A-curves for $n < 3$, a and b become infinite. For

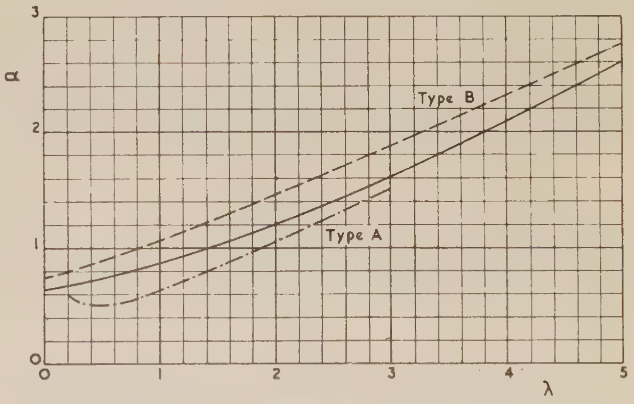


FIG. 10 CURVES REPRESENTING α AS A FUNCTION OF λ

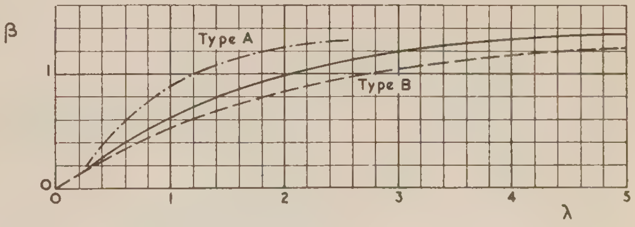


FIG. 11 CURVES REPRESENTING β AS A FUNCTION OF λ

small λ the best values of α and β depend very much upon the shape of the response curve near the origin. For these cases our theory would not be of interest anyway, as the values of our parameters become so high that on-and-off regulation would be the most favorable type of regulation that could be realized practically.

Final Choice of α and β

In view of the fact that a considerable deviation from the best values of the regulator parameters does not affect the quality of our regulation materially, the agreement of the α and β curves of both types may be considered fairly satisfactory. The response curves encountered in practice will have, roughly, the same shape as the curves of Types A and B. For any observed curve H, L, and L_2 or R_1 can be measured and thus λ is known. The corresponding optimum values of α and β will be somewhere near the values obtained for the Type A and B response curves. Owing to the relatively wide latitude of variation allowed for α and β we may assume universal values which we select somewhere intermediately between the two sets already found. These finally recommended values are recorded in Table 7 (for graphical representation see Figs. 10 and 11).

TABLE 7 SELECTED UNIVERSAL VALUES OF λ, α, AND β

λ	α	β	λ	α	β	λ	α	β
...	...	...	1.0	0.87	0.62	2.0	1.20	1.00
...	...	...	1.1	0.90	0.67	2.2	1.28	1.05
0.2	0.68	0.14	1.2	0.93	0.71	2.4	1.36	1.10
0.3	0.70	0.21	1.3	0.96	0.76	2.6	1.45	1.14
0.4	0.72	0.27	1.4	0.99	0.80	2.8	1.53	1.17
0.5	0.74	0.33	1.5	1.02	0.84	3.0	1.62	1.20
0.6	0.76	0.40	1.6	1.06	0.88	3.2	1.71	1.23
0.7	0.79	0.46	1.7	1.09	0.91	3.4	1.81	1.25
0.8	0.81	0.52	1.8	1.13	0.94	3.6	1.90	1.27
0.9	0.84	0.57	1.9	1.17	0.97	3.8	2.00	1.28
1.0	0.87	0.62	2.0	1.20	1.00	4.0	2.10	1.30

For λ > 3.5 the following formulas may be used

$$\left.\begin{aligned}\alpha &= 0.5\,\lambda + 0.1\\ \beta &= 1.6 - \frac{1.2}{\lambda}\end{aligned}\right\}\ldots\ldots\ldots\ldots\ldots\ldots[9]$$

It should be observed that these values of α and β should be used only for process-response curves of more or less "normal" shape. In exceptional cases one may meet with process-response curves which differ considerably from the hypothetical curves of Types A and B. For these cases the best values of the regulation parameters should be determined by separate investigation.

Practical Rules

Summarizing, we have the following practical rule: Find L, K, and H as indicated in Fig. 9, from the response curve of the process to a sudden increase q_0 of the regulating flow; calculate the lag quotient

$$\lambda = \frac{K}{H}$$

find α and β from Table 7, and take

$$\text{Sensitivity } a = \frac{\alpha}{K}$$

$$\text{Reset rate } = \frac{b}{a} = \beta/L$$

From the sensitivity, the proportional band (PB) may be computed easily.

$$\mathrm{PB} = \frac{1}{a}\cdot\frac{q_m}{x_m}\cdot 100$$

where

q_m = maximum interval of regulating flow
x_m = scale interval of regulated variable

The values of H, K, L, α, and β need be known only with an accuracy of, say, 10 per cent, because even a much greater variation in the adopted values of a and b does not make regulation materially worse.

It will be seen that the numerical values of the factors a, K, and H depend on the units of the variables q and x, whereas the b-values depend on the unit of time.

Processes Without Self-Regulation

For processes without self-regulation the parameter λ becomes zero, and the parameters α and β may be defined in the same way as before by Equations [7] and [8]. The values of α and β calculated from the data for Types D (Table 3) and E are given in Table 8.

TABLE 8 VALUES OF α AND β CALCULATED FROM DATA FOR TYPES D AND E

n	α	β
2	2.42	0.151
3	1.71	0.176
4	1.50	0.186
∞	1.11	0.22

Table 8 shows that there may be a considerable variation of the best values of a and b for different response curves possessing the same values of K and L. If we take as a reasonable compromise $\alpha = 1.5$, $\beta = 0.18$, we have the following practical rule for processes without self-regulation:

Find L and K from the response curve of the process and take

$$\left.\begin{aligned}\text{Sensitivity } a &= \frac{1.5}{K}\\ \text{Reset rate } \frac{b}{a} &= \frac{0.18}{L}\end{aligned}\right\}\ldots\ldots\ldots\ldots\ldots[10]$$

Furthermore, the same observations which have been made with respect to processes with self-regulation apply here also.

Finally, we wish to stress the following points:

1 Our theory starts from idealized assumptions, viz., the validity of the superposition principle both for process and regulator, and ideal proportional and integral action of the regulator. Either of these assumptions is only realized to a rather rough approximation.

2 When testing our theory on actual processes this should be done with respect to our criterion of minimizing the integral S. It goes without saying that the adoption of any other criterion leads to other optimum values of a and b.

Acknowledgment

Our acknowledgments are due to the Management of the N. V. de Bataafsche Petroleum Maatschappij (Royal Dutch/Shell Group) for its kind permission to publish this paper. We would also mention Messrs J. van der Veen, F. C. L. van Vugt, and E. Wieringa of this company, whose comments and criticisms have had considerable influence on the final shaping of this paper, and Mr. H. J. Krajenbrink, who carried out most of the computations.

The Exhaust-Heated Gas-Turbine Cycle

By D. L. MORDELL,[1] MONTREAL, CANADA

The author describes and discusses a gas-turbine cycle in which the combustion chamber is placed behind the turbine, raising the exhaust temperature to a level at which all the heat required for the compressor air can be supplied by the heat exchanger. The possibilities of such a cycle for the utilization of low-grade solid or liquid fuels are explained and attention is drawn to the convenience with which steam can be generated from the otherwise wasted heat, and the over-all thermal efficiencies which may be obtained by a combination of gas and steam turbines.

Introduction

In the past decade a great deal of experimental work has been done on the gas turbine, and much has been published on the subject. Surveying the present position, the advantages of the gas turbine, for fields other than aircraft propulsion, appear to be its compactness, low cost, and ease of control. With present metallurgical and aerodynamic limitations the simple "open cycle," with no allowances for auxiliaries, can achieve thermal efficiencies of the order 25 per cent, or, by the use of heat-exchange apparatus, 30 per cent. To improve compactness and specific weight, and to obtain a flat efficiency-load curve, resort may be had to compounding, intercooling, and reheating, which also will improve the thermal efficiency.

Criticisms which may be leveled at open-cycle gas turbines are that compared with a steam plant the efficiency is about the same, but the gas turbine demands more expensive fuels, as it is difficult to burn coal; and that, compared with the Diesel engine, although it can burn cheaper fuel, the gas turbine cannot yet approach the best efficiency of the Diesel.

By employing a closed cycle, the gas turbine can be made more compact and able to burn a wide range of fuels, including coal, but the engine becomes much more complex. There is no marked effect on thermal efficiency, which, as in the open cycle, depends on temperature, component efficiencies, and cycle layout.

It is the purpose of this paper to discuss a simple cycle in which solid and other low-grade fuels can be burned with a thermal efficiency of the order 20–25 per cent, excluding auxiliaries, and further to show how this may be brought up to nearly 40 per cent without departing from solid-fuel combustion, and without increasing the complexity of the plant unduly.

The Exhaust-Heated Cycle

The objection to burning solid fuels, and also certain low-grade liquid fuels, in an open-cycle gas turbine, lies in the difficulty of separating completely the ash from the combustion products. Ash passing through the turbine blades has a powerful erosive effect, while the accidental passage of any large pieces would be catastrophic. Furthermore, the deposition of solid matter on the blades has a deleterious aerodynamic effect that will demand frequent cleaning, necessitating opening the turbine. Residual fuel oils also may contain metallic salts which, after oxidation, may condense on the turbine blades, with serious effects.

In the closed cycle, the compressed air is heated in a form of air heater, so that the working fluid is quite distinct from the combustion gases. The fluid passing through the turbine is pure air, while any deposition of solid matter from the combustion gases is confined to stationary heat-transfer surfaces, from which it may be removed easily, if necessary, even while the plant is running.

In what the author terms the exhaust-heated cycle, the air used for combustion is that rejected from the turbine, and heating of the air prior to entering the turbine is accomplished entirely by a heat exchanger. This is illustrated diagrammatically in Fig. 1. No detailed discussion of this type of cycle has been found by the author in the published literature, although the principle is disclosed in patents of Haverstick[2] and Lysholm.[3]

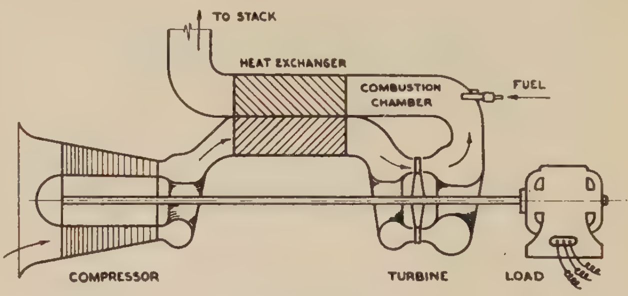


Fig. 1 Diagram of the Exhaust-Heated Gas-Turbine Cycle

It is evident that in such a cycle, compared with a conventional open cycle operating at the same compression ratio, turbine temperature, component efficiencies, and so forth, the transposition of the combustion chamber will not affect the specific power output (power developed per pound of air circulating per second), and, if the exchanger had 100 per cent effectiveness or thermal ratio, would not alter the thermal efficiency. For any thermal ratio less than 100 per cent, however, the final discharge temperature obviously will be higher than in the open cycle, leading to a reduction in thermal efficiency.

Cycle Performance

In calculating cycle performance, the basic assumptions made are as follows: Compressor efficiency, 85 per cent; turbine efficiency, 87 per cent; system pressure losses, 10 per cent; combustion efficiency, 100 per cent.

It is appreciated that these figures may be considered optimistic or conservative, depending upon the particular circumstances, but it was felt that they represent a reasonable basis for a general survey, while any particular value for the combustion efficiency simply multiplies the thermal efficiency.

Fig. 2 shows values of thermal efficiency and specific power for various values of the heat-exchanger thermal ratio for an assumed turbine-entry temperature of 1300 F. It is at once evident that thermal ratios of the order 0.75 are necessary for reasonable efficiencies, and the accompanying curves are based on this value.

---

[1] Associate Professor of Mechanical Engineering and Director of the Gas Dynamics Laboratory, McGill University. Mem. ASME.

Contributed by the Gas Turbine Power Division and presented at the Fall Meeting, Erie, Pa., September 28–30, 1949, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 49—F-6.

[2] U. S. Patent No. 2,438,635; J. S. Haverstick, March 30, 1948.

[3] "A Contribution to the Solution of the Gas Turbine Problem," by A. J. R. Lysholm, Proceedings of The Institution of Mechanical Engineers, vol. 157, 1947, pp. 498–513.

Fig. 3 shows the effect of turbine-entry temperature upon the performance. When using a heat exchanger, intercooling is usually beneficial, and Fig. 4 shows the effect of using an intercooler and a two-stage compressor. One stage of intercooling was assumed, placed so as to give equal temperature rises in each stage of the compressor. The improvement both in thermal efficiency and in specific power is quite considerable, and to complete the picture Fig. 5 shows the effect of reheating. This is accomplished as the air passes from the compressor turbine to the power turbine. Table 1 compares significant results. In each case we have assumed a heat exchanger of 0.75 thermal ratio and 1400 F turbine temperature.

The use of cooled blades in the turbine, permitting much higher gas temperatures, has not been presented at this stage. The benefits to specific output and efficiency are of course considerable, but their use is not considered practical in the immediate future because of the heat-exchanger tubes. For example, at 10 to 1 compression ratio, and turbine-entry temperature 2040 F, the specific output is 163 bhp per lb per sec, and the thermal efficiency 29.8 per cent, but the combustion temperature required would be 2500 F, with metal temperatures at the hot end of the exchanger tubes of 2240 F.

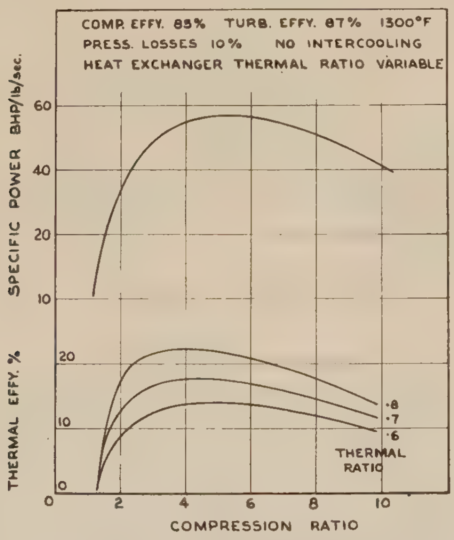


Fig. 2 Thermal Efficiency and Specific Power as a Function of Thermal Ratio and Compression Ratio

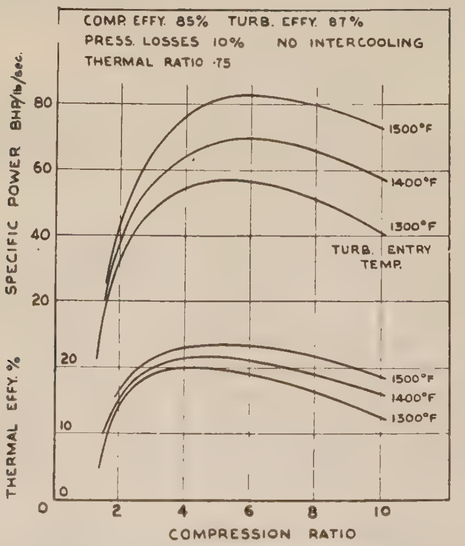


Fig. 3 Thermal Efficiency and Specific Power as a Function of Maximum Temperature and Compression Ratio

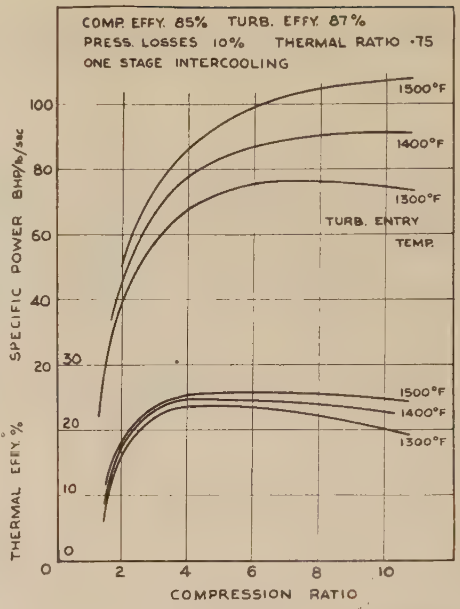


Fig. 4 Effect of Intercooling in Compressor

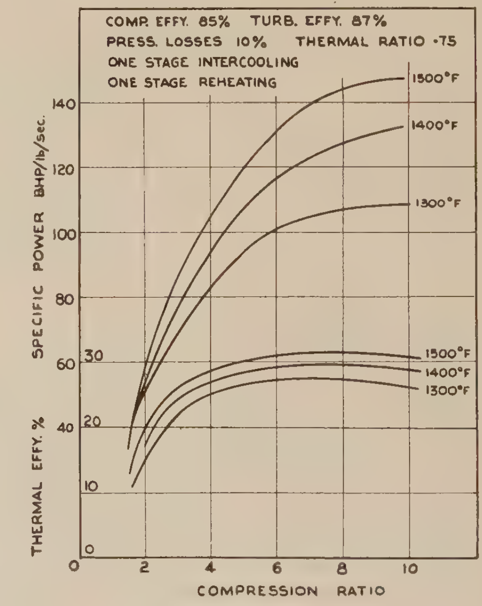


Fig. 5 Effect of Intercooling Compressor and Reheating in Turbine

TABLE 1 RESULTS OF PERFORMANCE ANALYSIS

	Compression ratio 5		Compression ratio 8	
	Specific output bhp/-lb/sec	Efficiency, per cent	Specific output bhp/-lb/sec	Efficiency, per cent
Simple open cycle	71.2	19.35	71.5	23.1
Open cycle with heat exchanger	69	28.5	65	26.6
Exhaust-heated cycle	69	22	65	19
Exhaust heat + intercooler	84	24.8	91	24
Exhaust heat + intercooler + reheat	108	28.5	127	30

Cycle Performance With Exhaust Steam Generation

The exhaust-heated cycle lends itself peculiarly well to the production of steam. The fact that exhaust temperatures are higher, and that the heat exchanger is at the exhaust end of the cycle means that it is only necessary to include extra transfer surface past which the exhaust will flow after heating the compressed air. The steam may be used as process steam, for district heating, or in a turbine coupled to the gas-turbine shaft. Aspects and estimates of the waste-heat boiler were recently discussed by Fusner.[4] The limitation on steam production is twofold. First we must not cool the exhaust below its dew point. This, however, owing to the large amount of excess air in the combustion chamber, will be in the region of 250 F, compared with the usual safe minimum temperature of 350–400 F used in boiler practice. The second factor is that in the production of the steam, a large fraction of the heat must be put in at, or above the saturation temperature, and this limits the amount of heat which can be extracted from the air by further cooling below the saturation temperature. Fusner based his estimates on the assumption that the air is cooled only down to the saturation temperature of the steam, so that his figures are somewhat conservative.

In Fig. 6 are shown the percentage heat transfer possible at various steam pressures and exhaust-gas temperatures, both for saturated and superheated steam. It should be emphasized that no account has been taken of dew-point considerations. The curves represent the thermodynamic limitations on heat transfer. The efficiency with which the steam is used also must be considered, and in Fig. 7 are shown the final results of using the steam produced by the waste-heat boiler in a steam turbine of 85 per cent efficiency and with a condenser pressure of 0.5 psi.

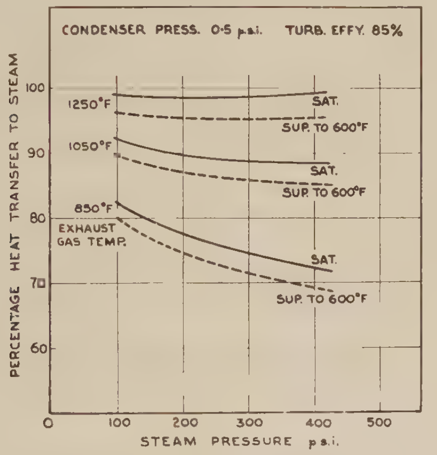


FIG. 6 FRACTIONAL HEAT RECOVERY IN A WASTE-HEAT BOILER

If we consider an exhaust-heated engine of compression ratio 4.5 to 1, with turbine-entry temperature of 1400 F, the thermal efficiency is 21.6 per cent. The exhaust temperature is just about 850 F. With this, using superheated steam, to avoid excessive moisture, we could recover 0.215 × 0.784 of the initial heat supply, or 16.8 per cent, giving an over-all efficiency of 38.4 per cent. The condenser and feedwater pump power demands probably would reduce this to about 36 per cent.

THE EXHAUST-HEATED CYCLE IN PRACTICE

So far we have considered only the thermodynamic aspects of exhaust heating. Summing up, exhaust heating gives a performance identical in specific output, and having about 75 per cent of the thermal efficiency of the conventional open cycle with heat exchanger. Let us now discuss the more practical aspects.

Combustion. The essential differences in the combustion chamber are that combustion is at an elevated temperature, practically atmospheric pressure, and the combustion products have to pass only through the heat exchanger. Thus we may expect a larger combustion chamber due to the lower pressure, but at the same time it is no longer a pressure vessel, permitting simplification in its construction. It is essentially a firebox similar to those in boilers, and as such, should be capable of burning any fuel that is ordinarily burned as boiler fuel. The elevated air temperature will assist materially in handling low-grade and high-moisture fuels. Since it is at almost atmospheric pressure, and could be at subatmospheric pressure were an induced-draft fan used to overcome the pressure loss in the heat exchanger, firing arrangements can be simple. One can envisage a hand-fired gas turbine without much trouble!

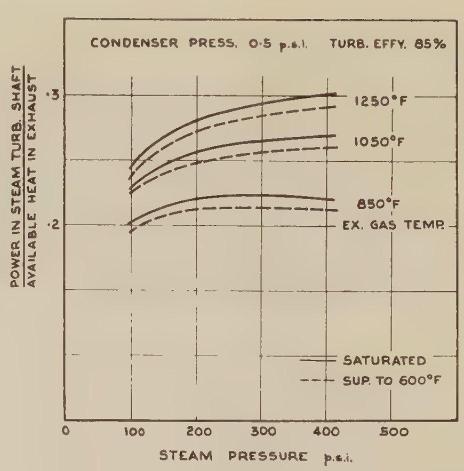


FIG. 7 EXHAUST-HEAT UTILIZATION IN A STEAM TURBINE

If a conventional mechanical stoker is used to burn coal, the undergrate air could not conveniently be taken from the main air stream owing to its high temperature, but it is a simple matter to provide a fan, which need be of very small power ($^{1}/_{2}$ to 1 per cent of the net output) to supply cool air under the grates. By the use of a suitable stoker and firebox design it would be possible to burn coal or oil indiscriminately.

The Heat Exchanger. The exchanger obviously is going to require 3 or 4 times the heating surface of the conventional exchanger, in addition to running at a higher mean temperature. A high thermal ratio is essential. There is one factor, however, which enables a given thermal ratio to be obtained by less surface area relative to its duty than would be required for the exchanger of a conventional cycle. Since we are heating the air immediately prior to its entry to the heat exchanger by an amount which will necessitate air-fuel ratios considerably greater than stoichiometric, we would normally separate the air for combustion and subsequent dilution. However, in the exhaust-heated cycle there is no need to obtain a uniform stream of gas for a turbine, and we conveniently may retain the separation, using the hot gases resulting from nearly stoichiometric combustion, for a second pass of the heat exchanger, the first pass being heated by the "dilution" air, which of course is already at a fairly high temperature.

The optimum choice of design here to obtain the smallest heat exchanger is complex and requires further analysis, but offers a prospect of a small saving in weight and bulk.

Cleaning of the heat-exchanger surfaces can be accomplished by soot blowers, while the engine is running, while ash or slag removal can follow normal practice.

Control. If a liquid or pulverized solid fuel is being burned, the power output of the engine may be controlled easily by altering the fuel-supply rate. If solid fuel is being burned on a grate, the simplest control would be, in effect, to by-pass the air from the firebox. This would lower the turbine-inlet temperature quite rapidly, depending upon the thermal capacity of the exchanger, while the hot fuel bed on the grate would remain, for quite a long time, at a temperature high enough to respond immediately as soon as air was readmitted under the grates.

---

[4] "The Gas Turbine With a Waste-Heat Boiler," by G. R. Fusner, *Mechanical Engineering*, vol. 70, 1948, pp. 515–518.

For starting up, on solid fuel, a supply of air under the grates would be required to light the fire and, when this was going well, the main compressor could be rotated by a suitable starting motor in the usual fashion. Due to the heat capacity of the exchanger, the motor would be required for a longer period than with the conventional engine, but need not be any greater in capacity.

Applications

The exhaust-heated engine can find application in any case where wide fuel utilization is important, and bulk and thermal efficiency are not prime considerations. It must be remembered that although the exchanger and combustion chamber are bulky, they must be compared with the equipment, such as gasifiers, or ash separators, or pulverizing systems that would be needed to burn solid fuel in the usual open-cycle unit. The thermal efficiency is only about 75 per cent of that of the conventional engines, but the price differential on the fuel consumable will be very much more than this, so that fuel costs will be considerably reduced. The first cost of the unit will be increased because of the larger exchanger, but in terms of cost per unit weight, the heat exchanger is about the cheapest component of the complete gas-turbine unit. The use of pure air in the turbines implies a reduction in maintenance costs, and this, together with the reduction in fuel costs, should more than make up for the extra initial cost, provided the utilization is high.

Electricity Generation. The combination of the exhaust-heated engine with steam generation, either for district heating, or extra shaft power, is most attractive. The thermal efficiency, at 36–38 per cent up, depending upon the auxiliaries, is obtained with few complications, and the plant consists essentially of a compressor, turbine, steam turbine, condenser, feed pump, vacuum pump, and combined combustor - heat exchanger - boiler.

The total transfer-surface area required will be about 2 sq ft per developed horsepower. The size of the combined heating unit for a 20,000-hp unit would be about 25 × 25 × 12 ft.

If the utmost efficiency is desired, Fig. 5 shows that thermal efficiencies of very close to 30 per cent are possible, by the use of intercooling and reheat. Reheating leads to higher final exhaust temperatures, making possible higher recovery factors. With a gas-turbine efficiency of 30 per cent and 25 per cent recovery of the waste heat, an over-all efficiency of 47.5 per cent is possible. The complexity of such a plant would be less than a mercury-steam unit, and the air is much easier to handle than the mercury vapor.

It would be practical, with but very slight modifications, to burn solid or liquid fuels or low-pressure gas. In locations where low-grade fuel is very cheap, economy may be sacrificed by using a cheaper and smaller heat exchanger of lower thermal ratio. The resultant higher exhaust temperature will permit increased utilization of the exhaust heat, so that even if the gas-turbine cycle efficiency drops to 10 per cent, the over-all efficiency of the dual-cycle plant can approach 32 per cent. Such a unit, in powers ranging up from, say, 4000 hp would be very attractive for some purposes.

Railroad Operation. Although the heat exchanger is large, it is not unreasonably so, and investigation has shown that there is enough room on a locomotive chassis to carry an exhaust-heated engine that is as powerful as a Diesel engine. A scheme for a 2000-hp unit is shown in Fig. 8. For railroad use the ability of the same engine to run, with but minor modifications, on coal or oil may prove to be a considerable advantage, while its ability to produce steam is of advantage for car heating.

Marine Service. In very large sizes, the exhaust-heated engine probably will be too bulky to compete with the fully closed-cycle engine. Its indifference to fuel type and its convenient and economic steam production, however, render it especially suitable for ships requiring power, say, less than 20,000 shp.

Conclusions

The transposition of the combustion chamber of a conventional open-cycle gas-turbine unit to a position between the turbine exhaust and the heat exchanger results in an engine arrangement, which, although bulkier and less efficient than the usual arrangement, possesses advantages entitling it to very serious consideration for many applications. The main feature of this exhaust-heated cycle is the fact that the turbines operate on pure air, and the combustion products pass only through the heat exchanger, thus dispelling any serious fears of ash troubles or turbine-blade erosion. A further advantage is the ease with which steam can be generated from the otherwise wasted heat, permitting high efficiencies where the steam can be employed usefully.

Starting and control appear to present no serious difficulties, whether the unit is run on low-pressure gas, lump coal, or low-grade crude oil, any one of which conceivably could be burned in the same engine.

Under circumstances where the relatively large bulk of the heat exchanger and combustor present no installation difficulty, the possibility of burning really low-grade fuels—a feature assisted by the high-temperature level of combustion—makes the exhaust-heated cycle, with or without exhaust steam generation, worthy of close study. In this paper we have attempted a brief review of the more obvious aspects, and hope that it may lead to a much closer and more detailed study of some of the problems suggested.

Acknowledgments

The author's thanks are due to Mr. W. A. Newman, manager of the research department, Canadian Pacific Railway, and Mr. J. I. Yellott, director of research, Locomotive Development Committee, for many valuable discussions and suggestions; and to Mr. A. G. Lane for assistance in the preparation of illustrations.

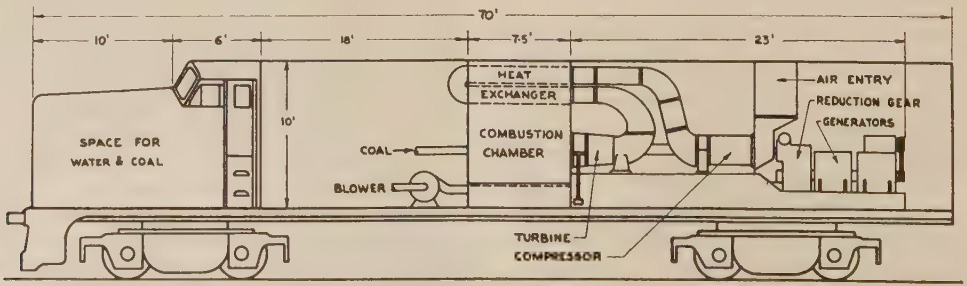


Fig. 8 Diagram Showing a 2000-Hp Exhaust-Heated Turbine in a Locomotive

Discussion

Frederick Nettel.[5] The basic arrangement which the author rightly recommends for many special applications is known from the time of the early air engines and was by no means overlooked when the gas turbine, or rather air turbine, created new possibilities in this field.

Since some relevant work was mentioned by reference to patents, it seems appropriate to draw attention to other work indicating new and substantial possibilities for improvement of the exhaust-heated gas turbine, by J. Kreitner and the writer.

As to Fig. 5 of the paper, it should be pointed out that the author evidently did not choose the optimum cycle which requires, after approximated isothermal compression, a temperature rise during compression

$$100 \times \frac{e(1-k)}{1-e(1-k)}$$

per cent of the absolute temperature at the start of the compression. This means that providing as the author does, for "equal temperature rise in each stage of the compressor" does not give best results. Nothing is said at what intermediate pressure reheat is applied. Optimal result requires a temperature drop during isentropic expansion, following approximated isothermal expansion, of

$$100 \times \frac{e\,(1-k)}{1-e{\cdot}k}$$

per cent of the absolute temperature at the start of the expansion. In these formulas e is the thermal over-all efficiency of the practical cycle, and k the effectiveness of the heat exchanger.[6]

The application of the optimal hexagon cycle would have resulted in higher efficiencies than those shown in Fig. 5, or the same efficiencies with a smaller heat exchanger and smaller intercooler.

Exhaust-Heated Cycle in Practice. It seems always advantageous to divide the air leaving the air turbine into two separate streams and to use the smaller of these part streams for burning the fuel with an air surplus as used in steam boilers. The second, larger part stream is used for preheating the air coming from the compressor and leaves the plant clean. It is thus available for many uses; for example, in the chemical industries, food processing, water distillation, and the like. Only about one third of the total heating surface requires periodic servicing.[7]

The air supply proposed by the author for stoker firing is not only wasteful but unnecessary. In order to get air of suitable temperature for the coal drier in a pulverized-coal-firing plant, or air for the various compartments of a traveling stoker, it is more advantageous to tap it from the said larger second part stream at one or more points within the heat exchanger where the exhaust air has been cooled by "useful" heat exchange with the compressed air coming from the compressor to a suitable temperature or temperatures.[8] This not only leads to considerable improvements in efficiency but also to better utilization of coal driers or of stokers, i.e., smaller stoker surfaces.

The exhaust-heated air turbine is by no means restricted to open cycles of comparatively small outputs, but can by modification to a semiclosed system be applied to plants of any size. In such high-pressure plants a turbopressure-charger set, handling only about one third of the air circulating in the main high-pressure circuit, precompresses the ambient air for the main circuit. The same fraction of the hot air leaving the main turbine drives the turbine of the charger set, and after leaving the latter, is used as combustion air in the "air boiler" at near-atmospheric pressure.[9]

Another interesting application of the exhaust-heated air turbine can be envisaged in the production of large quantities of more or less heated compressed air, for example, for metallurgical furnaces. Here the air can be simultaneously sharply dried practically without additional cost, and the danger of surging of the compressor, generally associated with large fluctuations in the air demand, eliminated.[10]

The combination of the exhaust-heated air turbine with steam generation is surely advisable where steam is required for process purposes, district heating, etc., but from the point of view of efficiency of size of plant it does not appear necessary. It seems too early to speak of costs, but the advantage of low cooling-water requirements would surely be lost in an air-steam plant. Besides, working with two heat carriers must at best be considered an unavoidable evil in certain cases.

The cited references should, in the opinion of the writer, show that there exist no basic difficulties in burning lowest-grade oils or coals in plants of sizes which may interest the public utilities for locations where a severe shortage of cooling water exists. Improvement in plant efficiency and increase in power output during the winter months should prove another attraction.

Many of the improvements described are applicable to locomotives and can be expected to improve substantially the efficiency and performance beyond that of the arrangement proposed in Fig. 8 by the author. The decision as to the chances of the exhaust-heated air turbine for rail traction will depend on whether or not the long-awaited pulverized-coal-fired locomotive with mechanical ash removal from the compressed combustion gases will prove a success.

From the foregoing it seems possible that the present preferred position of the closed cycle for large marine plants may eventually be challenged.

We may thus conclude that the prospects for the exhaust-heated air-turbine plants are even brighter than pictured by the author.

D. G. Shepherd.[11] The author has offered a clear and concise discussion of the exhaust-heated gas-turbine cycle, a topic which as he says, has been mentioned from time to time, but has not received detailed attention.

From the practical point of view, the crucial problem is that of the heat exchanger or "air boiler" which has to handle hot gas at an entry temperature of about 1725 F, air entering at about 420 F and leaving at 1400 F. (Figures based upon the author's assumptions for a 75 per cent exchanger and 1400 F turbine-entry temperature.) The use of a counterflow arrangement involves a tube temperature intermediate between 1400 F and 1725 F and therefore it may be necessary to utilize parallel flow for the first part of the exchanger, whereby the metal temperature may be reduced to an intermediate value between 420 F and 1727 F. Under these conditions, the duty would seem comparable to that of known superheater practice, with the advantage of much lower stresses owing to the lower pressure. Some advantage in size over the conventional heat exchanger for turbine cycles can be gained from the fact that the heat exchanger is in reality an air boiler, and a considerable part of the necessary heat transfer may be effected by radiation.

---

[5] Consulting Engineer, Manhasset, Long Island, N. Y. Mem. ASME.

[6] See also U. S. Patents 2,407,165, and 2,407,166.

[7] See U. S. Patent no. 2,394,253.

[8] See U. S. Patents nos. 2,434,950 and 2,420,335.

[9] See U. S. Patent no. 2,472,846.

[10] See U. S. Patent no. 2,457,594.

[11] Assistant Professor of Mechanical Engineering, Cornell University, Ithaca, N. Y. Jun. ASME.

The writer has found some difficulty in arriving at the figure of 22 per cent efficiency quoted in Table 1 of the paper for the exhaust-heated cycle of 5 to 1 pressure ratio, and it seems rather high. Possibly the difficulty lies in the evaluation of pressure losses, which are quoted as 10 per cent. Does this mean a loss of 10 per cent of the maximum absolute pressure, i.e., 7.35 psi, and if so, how is this loss apportioned before and after the turbine expansion? Perhaps the author will give us in more detail his basic assumptions in calculating the cycle performance.

For some applications, it will be more convenient to have a separate power turbine whose speed can be changed from that of the main shaft. This does not alter the conclusions of the author but does lend itself to some variants of the cycle. Thus the power turbine could have a separate combustion chamber with separate air supply bled off from the compressor. For the cycle mentioned, the work ratio is approximately $^1/_3$, so that with the same turbine temperature and pressure-loss ratios, the air would be split such that 2 parts go through the exhaust-heated side, which would act as a gas generator, and 1 part would provide the useful power. The power-turbine side would act on the conventional cycle, requiring fuel suitable for combustion before the turbine, with the gases from the turbine either passing through a separate exchanger or mixing with the main gas stream (as they have approximately the same exhaust temperature). This arrangement is illustrated in Fig. 9 of this discussion. Such a cycle gives an efficiency intermediate between the plain exhaust-heated cycle and the plain open cycle, roughly in proportion to the amount of air by-passed to the power-output side. It does of course depart from the cycle as described, since it cannot be run entirely on solid fuel, but has the advantage of more flexibility of operation, as well as providing an intermediate fuel economy which may be useful, depending upon the ratio of fuel costs.

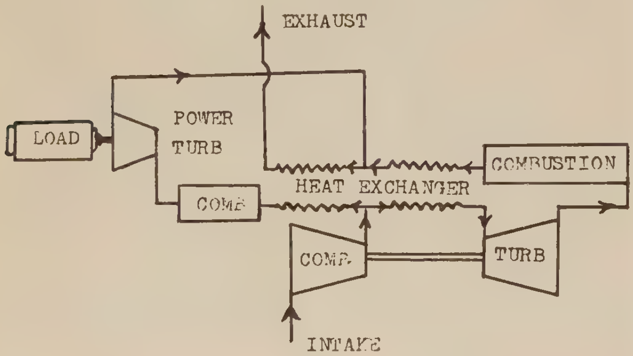


Fig. 9

R. A. Tyler[12] and D. C. MacPhail.[12] Professor Mordell has presented an admirable introduction to an interesting gas-turbine cycle. This cycle has been the subject of a preliminary examination in these laboratories with the specific application in mind of a coal-burning power plant for locomotive use of about 5000 hp output. In a power plant of this size the air heater assumes proportions significant to the practicability of the system and various types of heater form have been investigated.

The locomotive application necessitates careful consideration of the choice of heater construction. Long service life, ease of cleaning, and maintenance are fundamental requirements. Gas-side pressure losses in particular must be kept to a minimum and volume and material requirements be of a reasonable order.

It is believed that these requirements will best be met by a conventional shell-and-tube exchanger employing tubes of relatively large diameter for a reasonably small number of tubes under conditions of low-pressure loss. This type of exchanger can be constructed to incorporate the ruggedness necessary for locomotive operation and preliminary sketches indicate that a two-pass arrangement employing tubes as large as 1 in. in diameter can be accommodated in a locomotive layout of seventy feet over-all length. At this diameter an over-all tube run of about sixty feet would be required for a thermal ratio of 0.75 with a heat-transfer area per shp of about 3.5 sq ft. In view of the corrosive conditions and high temperatures prevailing in the shell it would appear impracticable to reduce the tube wall thickness below about 0.040 in. for a reasonable life expectation and the total weight of air heater would be about 10 lb per shp.

Stable operation and control of this form of power plant would appear to depend on a positive metering of a portion of the turbine exhaust to the firebox to effect a controlled rate of heat release, the metering being possibly effected by a simple valve of the constant-pressure type manually controlled from a "throttle" lever in the cab. This primary air in the form of hot furnace gas would be finally consumed in a separate mixing chamber with the remainder of the turbine exhaust, the mixed gases of uniform temperature proceeding to the shell of the air heater.

This physical separation of the primary and secondary air would appear desirable also from the point of view of grate-cooling requirements. The admission of fresh cooling air into the firebox must be kept to a minimum to avoid loss of over-all thermal efficiency through the rejection of this air at stack temperature. The throttle-controlled primary air entering the firebox would constitute about 25 per cent of the total turbine exhaust. The admission of cooling air into the primary air line could be effected by an automatic control of the cooling blower based on the limiting temperature (about 400 F) imposed by the grate. Under these conditions the reduction in over-all thermal efficiency due to cooling-air admission would be about 4 per cent, ignoring the negligible power requirements of the blower. The importance of controlling the cooling-air admission to a minimum level is illustrated by the extreme case in which the whole of the turbine exhaust is cooled to the limitation temperature. Under these conditions the over-all thermal efficiency would be of the order of 6 per cent.

The restriction of the firebox air to that required for the necessary heat release and the provision of a mixing chamber before the heater should result in a reduction of ash carry-over and a uniform temperature into the shell with consequent improved heater life. Further, with careful attention to the mixing-chamber design, a reduced pressure drop through the combustion system appears likely.

Author's Closure

Mr. Nettel has made many valuable suggestions for the proper application of the exhaust-heated cycle and it is obvious that he and his associates have been engaged on the close and more detailed study referred to in the conclusions.

In presenting performance calculations no attempt was made to find the optimum performance as such, we merely showed what could be done using reasonable compression ratios and maximum temperatures. For a given compression ratio equal division of the temperature rise between two stages gives the best compromise if the intercooler effectiveness be nearly unity, while in the use of reheat, practical convenience would make its application between the compressor and output turbine almost mandatory.

The suggestion to use some cooled "dilution" air to supply under the grates, rather than the separate supply envisaged by the author, is particularly valuable. The author shares with Mr.

---

[12] National Research Council, Ottawa, Ontario, Canada.

Nettel a dislike of two heat carriers, but it seems that the rewards would make it attractive so long as water is available.

In answer to Professor Shepherd, the effect of pressure losses in the system was allowed for by assuming the expansion ratio to be 90 per cent of the compression ratio. There is thus no need to specify where the losses actually occur at this stage.

In presentation of the paper, the author eschewed discussion of all but the simplest possible engine in the interests of conciseness, but it is apparent that there are innumerable variations of more or less complexity, to suit special needs. Professor Shepherd's proposed compromise appears to the author to abandon the one selling point of the true exhaust-heated engine, namely, its ability to burn very low-grade fuels.

The possibilities of the exhaust-heated cycle obviously are going to depend on the size of a practical heat exchanger, and the comments of Mr. Tyler and Dr. MacPhail are particularly timely.

A recent report by Kays and London[13] also studies the design problems of a heat exchanger for this application, and it is of interest to compare the two solutions. Messrs. Tyler and MacPhail have considered a very rugged design whose weight comes out at 10 lb per shp, and surface area 3.5sq ft/ shp. Messrs. Kays and London have examined four cases using smaller tubes and in some cases extended surface area. The analysis suggests weight of the order 2.4–5.6 lb per shp with volumes between 67.5 and 86.3 cu ft per 1000 shp. The optimum choice of type awaits trials in service, although it would appear necessary for railroad application to use the heavier design in which, presumably, maintenance would be cheaper and simpler.

---

[13] "Gas Turbine Regenerator Design Studies," by W. M. Kays and A. L. London, Technical Report No. 8, Department of Mechanical Engineering, Stanford University, Calif., 1949.

Properties and Characteristics of Fuel Oils for Industrial Gas-Turbine Usage

By D. P. HEATH[1] and E. ALBAT,[1] PAULSBORO, N. J.

Largely as a result of the rapid wartime development of gas turbines for aircraft usage, renewed emphasis has been placed upon the development of gas turbines for industrial applications. At the present time a number of domestic organizations are quite active in this field. One of the principal industrial applications of gas turbines in the United States is their use in the Houdry catalytic-cracking process in the oil-refining industry. Recent development work is centered around their use in the field of locomotive and marine propulsion using either coal or oil as fuel. The ultimate choice of a fuel is a complex problem involving such factors as fuel cost, engine performance and maintenance, and fuel availability. As far as fuel oils are concerned, the designer is further confronted with a wide variety of grades from which to choose. This paper presents information on the fuel properties and characteristics which are commonly employed for the identification and specification of such fuel oils, and also on those properties which are felt to be of significance with regard to fuel-system design and combustion-chamber and gas-turbine performance. Since fuel cost and availability are intimately related to supply and demand considerations, only limited regard is given to these factors in the present paper.

Typical Properties of Fuel Oils

THE classification of fuel oils into various grades has been tried several times with only limited success. A great many differences still exist among the systems employed by various oil companies; in addition, in many cases, there are considerable differences among the fuels produced by the different refineries of one oil company. A summary of several fuel-oil classifications is given in Table 1. Three different fuel-oil classification systems are presented in this table, ASTM Burner Oils, Pacific Coast Burner Oils, and ASTM Diesel Fuels. Of these three systems the ASTM Burner Oil Classification, which also has been adopted by the National Bureau of Standards, is the most widely used.

Average properties of the fuel oils produced in the United States are given in Tables 2, 3, and 4. In these tables the average properties and the normal variation in these properties are given for four types of distillate burner fuels, three types of residual burner fuels, and for three types of Diesel fuels.

The properties of the following four distillate-burning fuels are given in Table 2:

(*a*) Kerosine.
(*b*) Light distillate fuel oil.
(*c*) Domestic fuel oil.
(*d*) Heavy distillate fuel oil.

The main usage of kerosine is in wick-feed lamps and stoves, and in pot-type domestic oil burners. The main requirement of a kerosine for these purposes is good burning quality, i.e., it should burn without causing objectionable odors, smoke, or deposits. To meet this requirement, kerosine must have a boiling range of about 300 to 500 F and be paraffinic in nature.

Light distillate fuel oil is a product that is very similar to kerosine in physical properties. However, it usually will not meet all of the rigid kerosine specifications. Light distillate fuel is used mainly in pot-type domestic oil burners. Both kerosine and light distillate fuel oil frequently are sold as light Diesel fuels.

Domestic fuel oil is the most widely used distillate fuel. It has a boiling range of about 400 to 700 F and is used in all types of domestic oil burners. To burn satisfactorily in most domestic oil burners, a fuel should have an ASTM 90 per cent point below 675 F, as fuels having appreciable amounts of material above 675 F tend to smoke and leave deposits in those burners of poorer design. In addition, domestic fuel oil should be noncorrosive and should not deposit gum or sediment in the burner, fuel lines, or fuel tank.

A few types of domestic oil burners can burn satisfactorily a fuel of slightly higher boiling range than domestic fuel oil. In some localities heavy distillate fuels are available for use in these burners. Heavy distillate fuels have boiling ranges of about 400 to 750 F and have somewhat higher heats of combustion on a volume basis.

The properties of light and heavy No. 5 fuel oils and No. 6 fuel oils are given in Table 3. These stocks all contain residual materials and are used as industrial and marine fuels. The chief physical requirement of these fuels is that they be sufficiently fluid to permit ready transfer by pumping. For this reason, the classification of fuels into light and heavy No. 5 and 6 types was made on the basis of viscosity. The No. 5 fuel oils can be pumped without heating, except during extremely cold weather. The No. 6 fuel oils usually require heating to considerably higher temperatures than No. 5 fuel oils.

Typical physical properties of light, medium, and heavy distillate Diesel fuels are given in Table 4. The two major requirements of Diesel fuels are ignition quality, as measured by cetane number, and volatility. In certain critical Diesel engines (high-speed) the more volatile fuels give less exhaust smoke and odor, and fuels of high cetane number promote easier starting and smoother operation. Inasmuch as Diesel engines of a wide variety of sizes and speeds are produced, the volatility and cetane-number requirements vary considerably.

Light Diesel fuel is of the same boiling range as kerosine. Frequently, to simplify supply and distribution problems, a product meeting the requirements of both water-white kerosine, and light Diesel fuel is produced and sold as a common fuel grade. Similarly, domestic fuel oil is sometimes sold as medium Diesel fuel.

Specifications for the three highest-quality fuels—kerosine, medium Diesel fuel, and domestic fuel oil—are quite rigid; hence the range of properties encountered among several samples of these fuels is much narrower than will be encountered with most other fuels. The properties of residual fuel oils, in particular, will vary over a wide range.

---

[1] Socony-Vacuum Laboratories, Division of Socony-Vacuum Oil Company, Inc., Research and Development Department.

Contributed by the Gas Turbine Division and presented at the Fall Meeting, Erie, Pa., September 28–30, 1949, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 49—F-8.

TABLE 1 COMPARISON OF FUEL-OIL CLASSIFICATIONS

	Fuel Oils					Fuel Oils				Diesel Fuel Oils		
Designation	U.S. Govt. (NBS) CS-12-48 ASTM D 396-48 T					Pacific Coast Standards				ASTM D 975-48 T		
	No. 1	No. 2	No. 4	No. 5	No. 6	No. 100	No. 200	No. 300	No. 400	No. 1-D	No. 2-D	No. 4-D
Date of specification	Aug. 16, 1948					May 20, 1947				Sept. 29, 1948		
Physical and chemical tests												
Gravity, °API, min.	35	26	-	-	-	-	-	-	-	-	-	-
Distillation, ASTM method												
10% evaporation deg F	420 max.	440 max.[a]	-	-	-	350 min. 420 max.	425 min.	-	-	-	-	-
90% evaporation deg F	-	675 max.	-	-	-	450 min. 550 max.	600 min.	-	-	-	675 max.	-
End point, deg F, max.	625	-	-	-	-	-	-	-	-	625	-	-
Flash point, deg F, min.	100[b]	100[b]	130[b]	130[b]	160	110[b]	150	150	150	100[b]	100[b]	130[b]
Flash point, deg F, max.	-	-	-	-	-	165	-	-	-	-	-	-
*Pour point, deg F, max.	0	20	20	-	-	-	-	-	-	20[c]	20[c]	20[c]
Water and sediment, % vol., max.	Trace	0.10	0.50	1.00	2.00[e]	0.25	0.5	1.0	2.0	Trace	0.10	0.50
Carbon residue on 10% Btms, % wt., max.	0.15	0.35	-	-	-	-	-	-	-	0.15	0.35	-
Sulphur, % wt., max.	0.5[d]	1.0[d]	-	-	-	-	-	-	-	0.5	1.00	2.00
Sediment by extraction, % wt., max.	-	-	-	-	0.50	-	-	-	-	-	-	-
Ash, % wt., max.	-	-	0.10	0.10	-	-	-	-	-	0.01	0.02	0.10
Viscosity, temp., deg F	-	100	100	122	122	-	100	122	122	-	-	100
Max.	-	40 SSU	125 SSU	40 SSF	300 SSF	-	55 SSU	40 SSF	-	-	-	125 SSU
Min.	-	-	45 SSU	150 SSU @ 100 F	45 SSF	-	35 SSU	25 SSF	60 SSF	-	-	45 SSU
Kinematic viscosity, temp., deg F	100	100	100	-	122	-	-	-	-	100	100	-
Centistokes, max.	2.2	(4.3)	(26.4)	(81 @ 122 F)	(638)	-	-	-	-	-	5.8	-
Centistokes, min.	1.4	-	(5.8)	(32.1 @ 100 F)	(92)	-	-	-	-	1.4	-	-
Corrosion	No gray or black discoloration	-	-	-	-	-	-	-	-	Pass	Pass	-
Cetane number, min.	-	-	-	-	-	-	-	-	-	40[c]	40[c]	25

---

[a] 440 F maximum for use in other than atomizing burners.
[b] Or legal, see specification.
[c] See specification for alternates.
[d] For use where low sulphur fuel is required.
[e] See specification for method.
* Plus signs should appear before the number 20 across this line.

TABLE 2 PROPERTIES OF DISTILLATE-BURNER FUEL OILS

	Kerosine		Light distillate fuel		Domestic fuel		Heavy distillate fuel	
	Average	Variation	Average	Variation	Average	Variation	Average	Variation
Gravity, deg API	42.5	41- 44	42.0	40- 43	32.0	28- 37	29.5	22- 34
ASTM distillation, deg F								
Initial boiling point	340	330 - 380	350	330 - 390	370	320 - 390	400	360 - 460
10% recovered	390	365 - 420	390	370 - 420	430	390 - 475	480	440 - 500
50% recovered	435	420 - 450	445	425 - 460	500	450 - 530	535	490 - 560
90% recovered	490	460 - 510	500	480 - 520	570	500 - 600	640	590 - 680
Final boiling point	530	500 - 550	540	510 - 625	620	560 - 650	680	650 - 730
Flash point, deg F (Tag)	135	125 - 155	135	130 - 160	...	...	...	...
Flash point, deg F (PM)	...	...	...	...	160	135 - 190	180	160 - 210
Viscosity, centistokes at 100 F	1.75	1.6 - 1.9	1.8	1.4- 2.2	2.4	3.0	3.9	3.3 - 4.6
SSU at 100 F	...	...	...	...	34	32 - 40	39	37 - 41
Thermo viscosity at 60 F	415	370 - 460	...	...	...	..	...	...
Pour point, deg F	– 40	—25 -—50	—30	—20-—45	—20	0-—40	+10	+25-—35
Aniline cloud point, deg F	150	145 - 155	150	135 - 155	135	120 - 155	130	95 - 155
Sulphur, weight per cent	0.06	0.01 -0.08	0.08	0.04 - 0.5	0.2	0.06 - 1.0	0.3	0.1 - 1.0
Copper strip corrosion, 1 hr at 212 F	Pass	...	Pass	...	Pass	...	Pass	...
Color, Saybolt	+25	+20-+30	+15	+10-+25	...	...	...	...
Color, ASTM	...	...	...	...	1½	1 - 3	4	2 - 8
Water and sediment, vol per cent	Nil	...	Nil	...	Nil	0 - 0.1	Nil	0 - 0.2
Carbon residue on 10 per cent residuum, weight per cent	0.01	0.0 -0.02	0.01	0.005-0.15	0.05	0-0.35	0.2	0.1 - 1.4
Cetane number	50	48 - 52	50	45 - 52	...	30 - 50	...	20 - 45

TABLE 3 PROPERTIES OF RESIDUAL BURNER FUEL OILS

	Light No. 5 fuel		Heavy No. 5 fuel		No. 6 fuel	
	Average	Variation	Average	Variation	Average	Variation
Gravity, deg API	20	10– 30	19	14– 22	15	7– 22
Flash point, deg F (PM)	200	170– 320	220	180– 240	270	160– 410
Viscosity, SSU at 100 F	130	60– 300	...	...	...	...
SF at 122 F	...	...	30	20– 40	90	45– 300
Centistokes at 100 F	27.5	10.5– 65	130	65– 200	480	260– 750
Pour point, deg F	+20	—10–+80	5	—15–+50	45	+15– 85
Sulphur, weight, per cent	1.0	0.2– 3.0	1.0	0.5– 2.0	1.2	0.7– 3.0
Copper-strip corrosion, 1 hr at 212 F	Pass	...	Pass	...	Pass	...
Color	Black	...	Black	...	Black	...
Water and sediment, volume, per cent	0.2	Trace– 1.0	0.3	0.05– 1.0	0.4	0.05–2.0
Carbon residue, weight, per cent	3	1– 10	6	3– 8	7	5– 13
Ash content, weight per cent	0.03	0–0.10	0.03	0–0.10	0.05	0.01–0.50

TABLE 4 PROPERTIES OF DIESEL FUEL OILS

	Light Diesel fuel		Medium Diesel fuel		Heavy Diesel fuel	
	Average	Variation	Average	Variation	Average	Variation
Gravity, deg API	42.0	42– 43	36.0	33– 41	28.0	22– 31
ASTM distillation temperatures, deg F:						
Initial boiling point	345	330– 390	370	350– 460	390	360– 430
10% recovered	395	380– 420	450	420– 510	470	460– 480
50% recovered	445	430– 460	510	465– 550	540	520– 560
90% recovered	500	490– 515	550	530– 675	660	590– 680
Final boiling point	535	525– 625	620	600– 740	720	650– 750
Flash point, deg F (PM)	135	130– 160	160	140– 220	185	160 - 200
Viscosity, SSU at 100 F	...	...	...	...	40	38– 43
Centistokes at 100 F	1.8	1.4– 2.0	2.7	2.1– 5.8	4.2	3.6– 5.2
Pour point, deg F	—30	+20–—40	0	+20–—15	0	—30–+25
Aniline cloud point, deg F	150	145– 155	150	140– 160	130	110– 140
Sulphur, weight, per cent	0.05	0.005– 0.5	0.2	0.04– 1.0	0.4	0.2– 2.0
Copper-strip corrosion, 1 hr at 212 F	Pass	...	Pass	...	Pass	
Color, Saybolt	+23	+16–+27	...	...	...	...
ASTM	...	...	1-1/2	1– 7	6	4– >8
Water and sediment, volume, per cent	Nil	...	Nil	Nil–0.10	0.01	0–0.50
Carbon residue on 10 per cent residuum, weight per cent	0.01	0–0.15	0.03	0.01–0.35	0.1	0.01– 0.2
Cetane number	50	40– 54	50	40– 53	30	25– 40

All of the ten classifications of oils listed in Tables 2, 3, and 4 are not available in any one locality. In most localities in the United States three to seven classifications of fuel oils will be available. The three fuels that are most widely distributed are kerosine, domestic fuel oil, and No. 6 residual fuel. Some changes in the availability of oil stocks can be expected from year to year, but residual fuels will continue to be the most plentiful, with domestic fuel oil being the most widely available distillate fuel.

Significance of Physical Tests

In studying various physical tests of fuel oils, it should be kept in mind that no one test taken alone is of much value, but that a group of tests is necessary before the characteristics of the oil can be evaluated. With this in mind, the significance of the various tests listed in Tables 1 through 4 are discussed briefly.

API Gravity. API gravity is a direct measure of the density of an oil and is a good indication of its heating value. The density of an oil changes rather rapidly with temperature. Fig. 1 is useful in estimating the density at various temperatures when the API gravity is known. API gravity is also an indication of the composition of an oil. For example, of two oils having the same boiling range, the one having the higher API gravity is the more paraffinic.

ASTM Distillation. An ASTM distillation is one of the most common tests conducted on petroleum products. It is a measure of the boiling range of an oil and, as such, is of different significance with different products. In domestic fuel oils the amount of high-boiling material present gives an indication of the performance of the oil in certain types of burners. Specific burner types are commonly designed to operate on fuels of a given volatility. Use of a fuel higher in boiling range than that for which the burner was designed usually will lead to increased smoke and deposits. For example, most gun-type household burners can burn successfully a heavier fuel than pot-type burners, while an industrial-type burner can burn practically any liquid fuel.

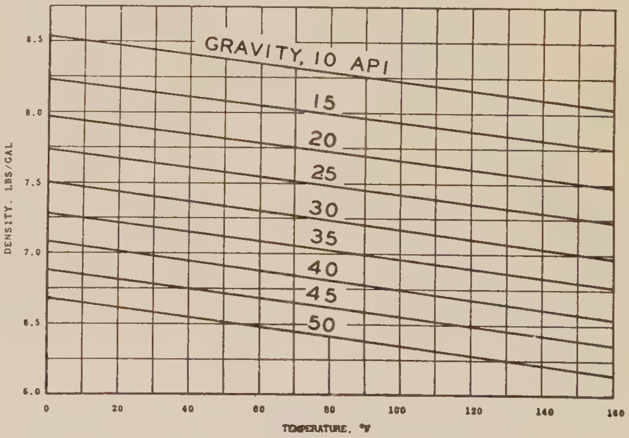


Fig. 1 Density Characteristics of Fuel Oils

Flash Point. The flash point of an oil is the lowest temperature of the oil at which the vapor above the oil will burn if subjected to an open flame. Therefore the flash point of an oil is an indication of the temperatures at which an oil can be handled without undue fire hazard. Both the Pensky-Martens and Tag flash points, listed in Table 2, are determined in a closed container and agree quite well with each other; however, the Tag flash point procedure should not be used for oils having flash points above 175 F. The flash point determined in a closed container is always a few degrees lower than a flash point determined in an open container.

Viscosity. Several different methods of measuring viscosities are used for fuel oils, each grade of fuel oil having a particular viscosity method that is most widely used in its specification as shown in Table 5.

Kinematic viscosity at 100 F is gradually replacing the various Saybolt viscosity measurements as it is a more accurate and more

TABLE 5 METHODS OF MEASURING VISCOSITY

Product	Type viscosity measurement usually specified
Kerosine	Saybolt thermo viscosity at 60 F
Diesel fuels Distillate fuel oils	Saybolt Universal or kinematic viscosity at 100 F
Residual fuels	Saybolt Furol viscosity at 122 F

flexible method of test. Kinematic viscosity at 100 F can be converted to the various Saybolt measurements by means of the scales given in Figs. 2 and 3. Typical viscosity-temperature curves for the various grades of fuel oil are also given in these figures. Since No. 6 fuel oils cover such a wide range of viscosities, two lines are given for this grade.

A lower limit of 30-sec Saybolt Universal at 100 F on Diesel fuels is generally recognized as being necessary to prevent leakage in worn fuel-injection equipment, and to furnish lubrication for the injection equipment of certain types of engines. For operation at atmospheric temperatures below 10 F, high-viscosity fuels are usually avoided to secure the required low pour point and increased volatility. If the viscosity of a Diesel fuel is unusually high, a solid stream may be obtained from the injector instead of a spray.

In the utilization of residual fuels in marine boilers, which usually do not have steam atomizers, an optimum viscosity of the fuel at the nozzles can be found which results in the most efficient combustion. Preheating of viscous boiler fuels to give a viscosity of about 150 sec Saybolt Universal at the sprayer plates is standard U. S. Navy practice.

Detailed relationships of the effects of viscosity upon spray characteristics of gas-turbine fuel nozzles are not available, but it is expected that the design of the atomizing equipment will provide no difficult problem.

Viscosity is also related to ease of pumping. In general, if an oil has a viscosity of over 5000 to 10,000 sec Saybolt Universal, it is extremely difficult to pump. Inasmuch as the viscosity of an oil decreases rapidly as the oil is heated, very viscous oils require heating before they can be pumped. The temperatures to which an oil of a given viscosity must be raised to facilitate easy pumping can be estimated by the use of Figs. 2 and 3.

Pour Point. The pour point of an oil is the lowest temperature at which an oil will flow under specified test conditions. It is related directly to the wax content and viscosity of an oil. In general, the lowest practical pumping temperature is considered to be about 10 deg F above the pour point.

Aniline Cloud Point. The aniline cloud point is a measure of the paraffinicity of an oil, a low aniline point indicating an aromatic, naphthenic, or a highly cracked oil. Conversely, a high

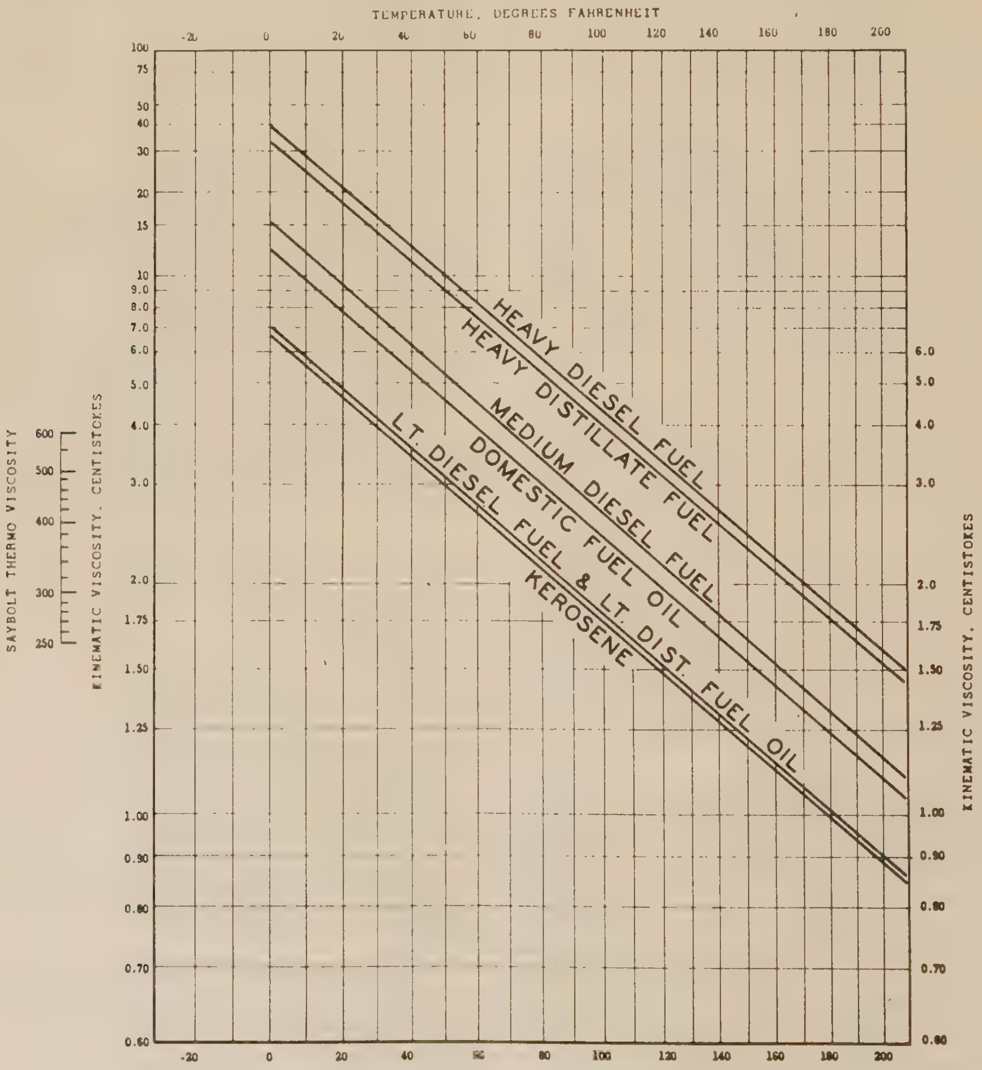


FIG. 2 VISCOSITY CHARACTERISTICS OF REPRESENTATIVE FUEL OILS

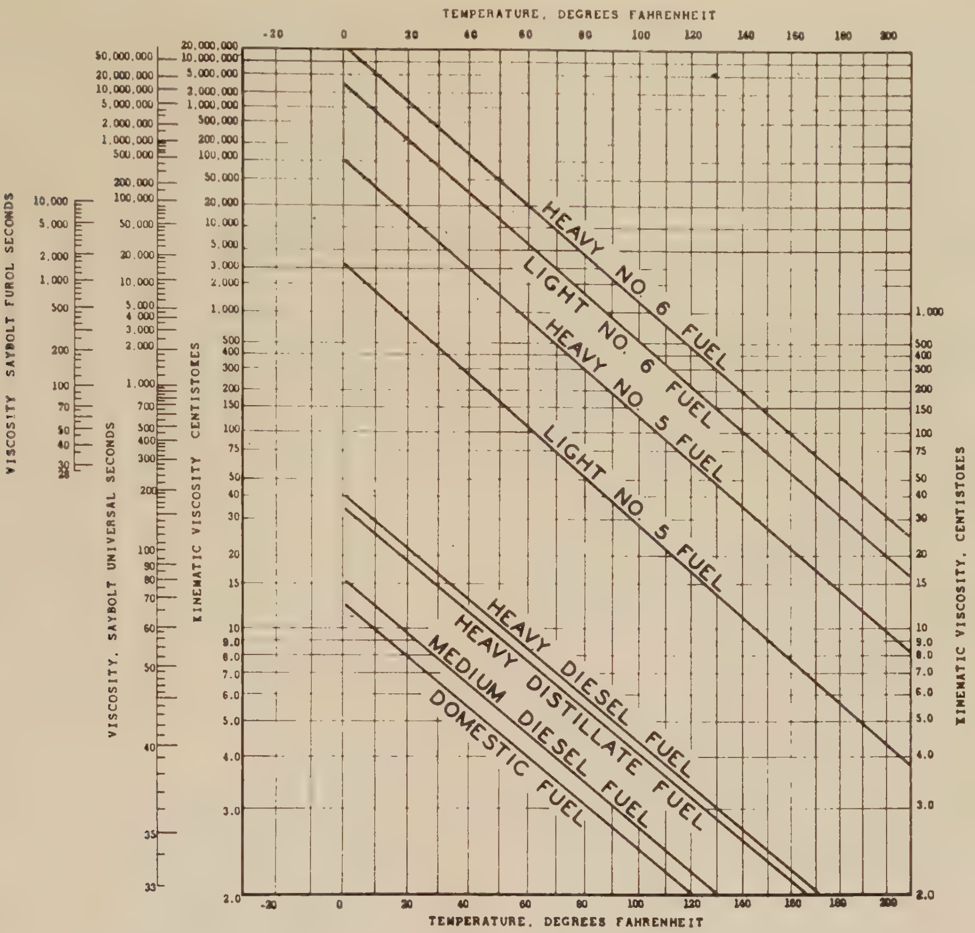


Fig. 3 Viscosity Characteristics of Representative Fuel Oils

aniline point indicates a paraffinic straight-run oil. The major use of the aniline point is in the calculation of empirical constants related to some performance characteristics. These constants are discussed in a later section of this paper.

Sulphur Content. From a corrosion standpoint, sulphur content is important only when it is excessive. It must be remembered that sulphur occurs in many forms of which some are harmful and some are not.

The sulphur content of kerosines, which are to be burned in lamps or stoves indoors obviously should be quite low because of the resultant odor of the products of combustion. However, the sulphur content of other burner fuels is not of importance from the standpoint of corrosion in the exhaust or chimney unless the products of combustion are cooled sufficiently to condense the water in them. Sulphur dioxide is not a corrosive agent except in the presence of water.

With respect to corrosion at high temperatures, of the order normally encountered in gas-turbine applications, Avery (1)[2] presents quantitative data showing the effects of sulphur-bearing combustion gases on various metal alloys. It was shown that even high-sulphur gases do not cause a measurable increase in corrosion rate over that which would be obtained with air of the same temperature, when high-temperature-type alloy steels are used.

Copper-Strip Corrosion. Copper-strip corrosion tests attempt to measure the corrosive tendencies of an oil. Usually such tendencies are related to the presence of harmful sulphur compounds. The results of corrosion tests are obvious for very bad fuels or very good fuels, but intermediate cases are not conclusive. The corrosion occurring in a system handling fuel oil is more frequently the result of the existing temperature-humidity conditions or the materials of construction rather than the result of operation on a corrosive fuel.

Color. The color of an oil is of little value as an indication of its performance characteristics except in extreme cases wherein contamination is indicated. The two common methods of measuring the color of petroleum oils are the Saybolt and ASTM methods. The Saybolt colorimeter is used on white products such as kerosine and gasoline. The Saybolt color scale runs from —16 (darkest) to +30 (lightest). The ASTM colorimeter is used to measure the color of fuel and lube oils. On the ASTM color scale, No. 1 is the lightest and No. 8 is the darkest color.

Water and Sediment. The presence of water and sediment is usually an indication of fuel-oil contamination. In the case of residual fuels, high values for this test may also be related to the instability of the fuel which is occasioned by the precipitation of heavy asphaltic or carbonaceous substances. The sediment-by-extraction test is another test frequently specified for heavy fuel oils and is supposed to give a similar indication of the insoluble material dispersed in the oil which ultimately will sludge out. Neither of these tests has been found to correlate with the preheater fouling characteristics of certain residual fuel oils. Presumably, polymerization and oxidation effects associated with the frequent recirculation of oil at relatively high temperatures overbalanced these factors. As a result, the Navy Department, in specifying stable residual fuel oil for its vessels, requires that it pass a heater-fouling test which simulates preheater operation aboard ship.

---

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Another well-known cause of sludge and sediment in fuel oils is that resulting from the incompatibility of mixtures. A gross example of this would be the mixing of a straight-run Diesel fuel with a deeply cracked bunker C fuel oil. Though neither fuel would have large amounts of sediment present, as indicated by the BS&W test, the mixture would have a very large amount which could plug lines, filters, and preheaters. Vessels are particularly susceptible to this type of fuel failure where bunker liftings from different countries are made. Where there is doubt concerning the compatibility of fuel oils, segregation of the stocks will of course minimize operating difficulties.

Ash Content. Residual fuel oils usually possess ash-forming constituents which indicate the presence of inorganic materials. A wide variety of metals may be found in the ash which originates in part from the metals inherent in the crude oil, the contamination by chemical compounds during the refining operation, and the entrainment of materials such as rust, seawater, etc. Sodium, iron, and vanadium are particularly deleterious to the brickwork in a furnace and cause rapid slagging of the refractory materials, necessitating frequent and costly replacement. In addition, these materials in the form of sulphates and oxides deposit on the closely packed superheater tubes and restrict the passage of the combustion gases through the furnace.

In so far as gas-turbine operation is concerned, a high ash content in the fuel oil may increase the rate of erosion of the burner chamber, turbine nozzles, and blading. In a recent test (2) with a high-ash-content fuel (1.5 per cent) no serious effects were noted. Efforts on the part of the refinery to reduce the ash in residual fuels have not been successful without a disproportionate increase in the cost of the fuel. Filtration and centrifuging were found to be partially effective, depending on the nature of the oil and the ash-forming constituents as well as the conditions of processing.

Carbon Residue. The carbon residue of distillate-type fuels is generally too small to be measured accurately. The carbon-residue test on the 10 per cent bottoms from the fuel is more reproducible and serves to indicate the presence of residual materials. Carbon residue has some relation to burner deposits caused by fuel oils and is felt to be related to combustion-chamber deposits in Diesel engines; however, no definite relationship has been found. In residual fuel oils, a high carbon residue indicates a deeply cracked fuel oil or, in some cases, residual materials from highly asphaltic crude. Paraffinic straight-run residual fuel oils are characterized by a relatively low carbon residue.

Heat of Combustion. The heat of combustion of an oil is usually estimated from the API gravity inasmuch as the actual determination is quite difficult. Fig. 4 shows the relationship between API gravity and the net or lower heating value. In general, the differences in heat of combustion between different batches of the same grade of fuel are too small to be reflected in engine or burner performance. However, in carefully controlled laboratory tests of Diesel engines, heat of combustion has been found to be a variable affecting power output and fuel consumption.

Cetane Number. Cetane number is of importance only in Diesel fuels. It is one of the factors affecting engine starting and roughness in operation, but appears to have little or no direct relation to exhaust smoke, power, or fuel consumption.

Heat Content and Related Thermal Properties. The thermal properties of fuels, such as heat content, heat capacity, and thermal conductivity, often must be estimated in the design of fuel-oil systems. Average values for these variables are given in Figs. 5 to 7, inclusive, which should prove useful to the design engineer.

Spontaneous-Ignition Temperature. A measure of the ease with which an oil will ignite in contact with a hot object in the presence of air is given by the spontaneous-ignition temperature. It is not entirely a fuel property, however, as the values will vary considerably depending upon the apparatus used and the method of test employed. As an indication of the values obtained, some of the results of Edgar (4) are given in Table 6.

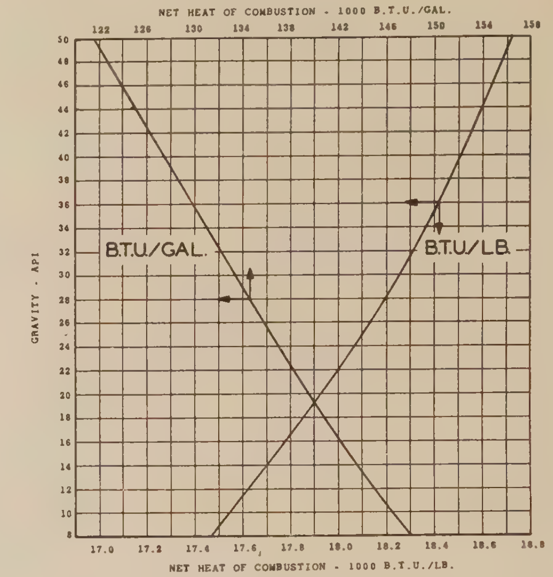


Fig. 4 Heat of Combustion of Petroleum Fractions at Constant Pressure

Surface Tension. Surface tension is the molecular force existing in the surface film of all liquids which tends to contract the volume into a form having the least surface area. Compared to water, petroleum products have relatively low surface tensions. Water has a surface tension of 76 dynes per cm at 20 C (68 F), while petroleum products have surface tensions varying from 15 to 40 dynes per cm at 20 C.

TABLE 6 SPONTANEOUS IGNITION TEMPERATURE OF PETROLEUM PRODUCTS (4)

Product	SIT, deg F
Aviation gasoline (100 octane leaded)	960
W.W. kerosine (fly-spray)	520
Diesel fuel (52 cetane)	790
Diesel fuel (48 cetane)	790
No. 1 fuel oil (Mid-Continent)	530
SAE 60 aero lube oil (MC solv. ext.)	750

This property in petroleum distillates appears to be a direct function of the density of the oil. In Fig. 8 the surface tensions of a number of pure hydrocarbons boiling in the gasoline range (5) are plotted against specific gravity. The line drawn in Fig. 8 is that given for petroleum distillates having end points up to 570 F (6).

The surface tension of petroleum products decreases with increasing temperature. From the limited amount of data available, it appears that all types of hydrocarbons have approximately the same rate of change of surface tension with temperature. In the range of 0 to 100 C (32 to 212 F), this rate of change is approximately 0.11 dynes per cm per deg C (0.061 dynes per cm per deg F).

In aqueous solutions, the addition of wetting agents or detergents causes an appreciable reduction in the surface tension of the liquid and gives a solution which will form a relatively stable froth with air. In petroleum products, however, the addition of antifoam agents or surface-active agents brings

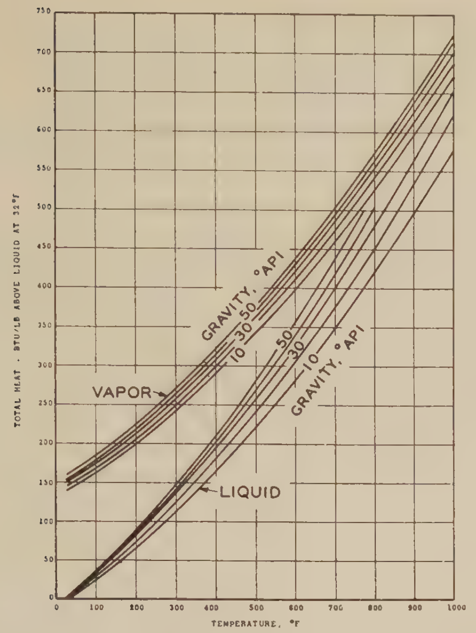


FIG. 5 TOTAL HEAT CONTENT OF PETROLEUM FRACTIONS (Average stock.)

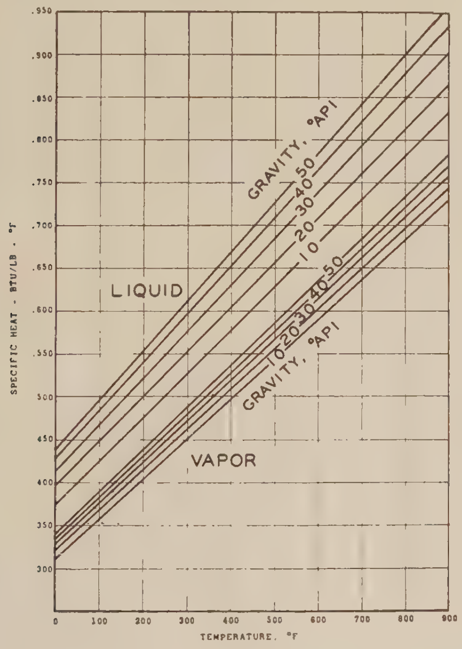


FIG. 6 HEAT CAPACITY OF PETROLEUM FRACTIONS (Average stock.)

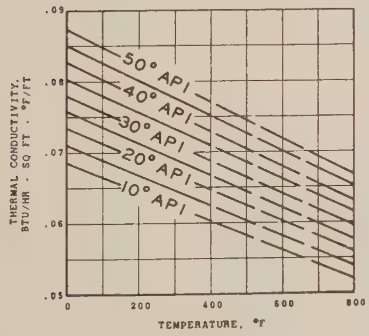


FIG. 7 THERMAL CONDUCTIVITY OF PETROLEUM LIQUIDS

about a reduction of less than 3 dynes per cm in the surface tension of the oil. Inasmuch as the surface tension of a petroleum distillate of a given boiling range will fall within a relatively narrow range, and this range can be changed only slightly by the use of addition agents, the surface tension of a fuel oil would appear to be a very minor variable affecting the characteristics of the fuel spray from a nozzle.

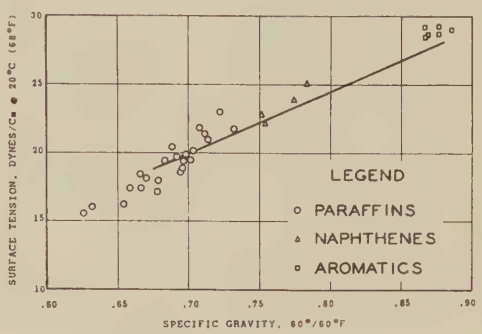


FIG. 8 SURFACE TENSION OF PETROLEUM OILS

OTHER CHARACTERISTICS AND TESTS

Empirical Constants. It is apparent that the performance characteristics of an oil are quite difficult to estimate from the usual physical tests. Since performance tests are quite difficult and expensive to make, several empirical constants utilizing two or more physical tests have been developed for estimating product performance. These constants usually employ two or more of the following properties: API gravity, viscosity, aniline cloud point, and average boiling point. Since all of these properties are interrelated, the use of two of them is usually sufficient to obtain an indication of the composition of an oil and hence of its performance properties.

Diesel Index. Diesel index is a relatively reliable index to cetane number. It is usually several numbers higher than the corresponding cetane number, except in the case of fuels containing additives to improve the cetane number in which case the relationship may be reversed.

The relationship between Diesel index and cetane number is about as follows

Diesel index	Cetane number
43	40
50	45
59	50

Diesel index is applicable only to fuels having a boiling range similar to that of Diesel fuel. For example, aviation gasoline has a Diesel index of about 70, and a cetane number of below 10.

Diesel index is calculated from the API gravity and the aniline cloud point by the following equation

$$\text{Diesel index} = \frac{\text{API gravity} \times \text{aniline point, deg F}}{100}$$

UOP K-Value. The UOP *K*-value or characterization factor has been widely used as an index of composition in the correlation of thermal properties, such as heat content and specific heat of oils. It has also been used to indicate the burning qualities of fuel oils. The range of values obtained is quite limited. Most oils will be between 10.0 and 12.5 with the paraffinic stocks having the higher values. The *K*-value is calculated from the average boiling point and specific gravity by the following equation

$$K = \frac{\sqrt[3]{T_B}}{S}$$

where

T_B = average boiling point, deg R = deg F + 460
S = specific gravity 60/60 deg F

Carbon-Hydrogen Ratio. The carbon-hydrogen ratio of distillate fuels has been found to correlate with the carbon deposits formed in domestic oil burners under certain operating conditions. Here again, the design of the burner is of primary importance and, in critical units, fuels having low carbon-hydrogen ratios, i.e., the more paraffinic fuels, will result in smaller deposits.

The carbon-hydrogen ratio of a fuel can be determined by burning a given amount of oil completely and analyzing the products of combustion. However, this procedure is extremely difficult to carry out accurately. A correlation for estimating carbon-hydrogen ratios of distillate fuels has been developed by Cauley and Delgass (7) and is presented in Fig. 8. This correlation is based upon aniline point, API gravity, and boiling point of a fuel (ASTM 50 per cent boiling point). The empirical formula expressing the relationship in Fig. 9 is as follows

$$\text{C/H ratio} = \frac{28}{\sqrt[3]{A} + 47}$$

Application of this correlation is limited to distillate fuel oils with mid-boiling points ranging from 350 F to 671 F.

PRODUCTION OF FUEL OILS

The production of fuel oils, in general, is controlled by refining operations for the production of more valuable products. Fuel oils, ranging from light domestic fuel and high-speed Diesel fuel, to heavy marine and industrial fuel are obtained as distillates, or as residuum from primary crude-oil distillation, or from thermal and catalytic cracking. All types of fuel oils can be converted to gasoline by thermal or catalytic cracking, assuming the availability of suitable processing equipment.

Since refining operations are ordinarily carried out primarily to obtain the desired yield and quality of gasoline, it is not possible to exercise complete control over the characteristics of the fuel oils produced simultaneously. In the case of crude-oil distillation, the light-fuel boiling range can be controlled, but not much can be done about any of the other characteristics. In other words, the refiner simply isolates a fuel of desired boiling range and its characteristics depend only on the type of crude processed. In the case of heavy fuels, the chief property which can be regulated is viscosity. This characteristic is controlled by blending light fuel-oil (distillate) fractions in the residual (undistilled) fuel to the extent required. Similarly, the light fuel produced by thermal and catalytic cracking can be controlled as regards boiling range, but the other characteristics depend primarily on the type of stock being processed and on the severity of the cracking operation. The viscosity of the heavy fuel produced through thermal cracking is regulated, as in the case of crude-oil distillation, by blending-in light fractions. Thus the production of fuels of the desired characteristics is mainly a matter of selection and blending of available stocks.

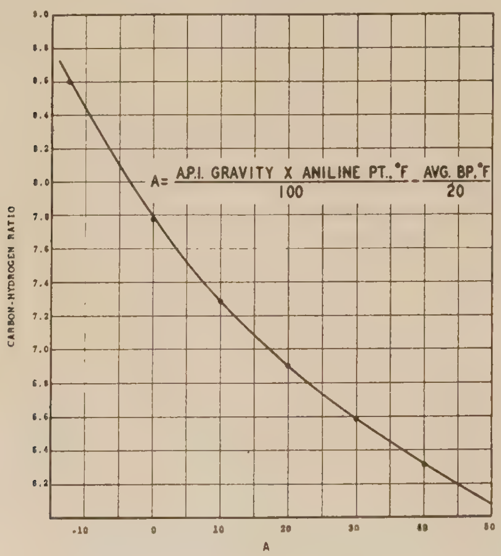


FIG. 9 CARBON-HYDROGEN RATIO OF DISTILLATE FUEL OILS

The stability and performance specifications for kerosine, Diesel fuel, and domestic heating oils limit the types of refinery stocks suitable for these fuels. In general, burning kerosines contain only straight-run material, light Diesel fuel contains straight-run and catalytically cracked fuel, and domestic fuel oil can contain straight-run, catalytically cracked, and thermally cracked fuel.

AVAILABILITY OF FUEL OILS

It is difficult to predict reliably what the future relationship of supply and demand for fuel oils will be. With the completion of refinery-expansion programs, and the modernization of transportation systems, supply has more than kept up with demand. The result has been a cutback in crude-oil production today. The fact remains that, when needed, enough oil will be produced to satisfy all essential requirements in the near future, despite the predictions of some forecasters to the contrary.

An estimate of the future fuel-oil balance can be made from current consumption trends. For 1947, Table 7 (8) gives the consumption of fuels in the United States. It is noted that heating and industrial applications were the principal consumers of fuel oils. Residual fuels constituted the bulk of the oils and averaged about 56 per cent of the total; distillate fuels came next and comprised about 25 per cent. Kerosine and Diesel fuel made up the remainder of 11 and 8 per cent, respectively. At the moment, domestic oil-burning installations are proceeding at an accelerated rate which, together with the steam-to-Diesel con-

TABLE 7 U. S. CONSUMPTION OF FUEL OILS FOR 1947 (8) IN THOUSANDS OF BARRELS OF 42 GAL

	Kerosine	Diesel fuel	Other gas oil and distillates	Residual fuel oil	Total	Per cent of total use
Heating..........	...	...	178359	56402	234761	25.5
Range oil.........	62482	...	11632	...	74114	8.0
Tractor fuel.......	8209	9079	4118	...	21406	2.3
Railroads.........	...	20476	3143	97500	121119	13.2
Industrial.........	...	10389	16291	177757	204437	22.2
Public utilities....	...	7100	7116	60964	75180	8.1
Vessels...........	...	17313	2338	121047	140698	15.3
Miscellaneous.....	32012	5500	5160	6859	49531	5.4
Total....	102703	69857	228157	520529	921246	100.0
Per cent of total fuel oil..........	11.1	7.6	24.8	56.5	100.0	

TABLE 8 ESTIMATED U. S. CONSUMPTION OF FUEL OILS IN THOUSANDS OF BARRELS OF 42 GAL

Fuel	1947	1951	Per cent change over 1947
Kerosine	102703	113000	+10
Diesel fuel	69857	113000	+62
Distillate fuels	228157	329000	+44
Residual fuels	520529	474000	−9
Total	921246	1029000	+12

version program of the railroads, and the increased use of Diesel tractors, buses, and trucks, will impose increased demands on distillate-type stocks. The extent of these increases for 1951 is estimated in Table 8.

A reduction in the amount of residual fuel oils available is predicted. However, they will still be the main fuel-oil type for some years to come. Part of the increase in distillate fuels will be accomplished at the expense of residual stocks which will be cracked more extensively. In the immediate future, considerations of supply point to the use of these residual grades as the most attractive liquid fuel for gas-turbine usage.

BIBLIOGRAPHY

1 "Estimation of Scaling Resistance," by H. S. Avery, Alloy Casting Bulletin, No. 10, May, 1947, p. 9.

2 "Combustion System for Burning Bunker C Oil in a Gas Turbine," by B. O. Buckland and D. C. Berkey, Paper No. 48—A-109, presented at the Annual Meeting, New York, N. Y., November 28–December 3, 1948, of The American Society of Mechanical Engineers.

3 "Thermal Properties of Petroleum Products," Bureau of Standards, no. 97, 1929, p. 24.

4 "Prospects for Use of Safety Fuels in Spark-Ignition Aircraft Engines," by Frank C. Mock, discussion by Graham Edgar, SAE Transactions, vol. 34, 1939, p. 294.

5 "Annual Tables of Physical Constants," Section 700 (C), 1, Frick Chemical Laboratory, Princeton, N. J., 1941.

6 "International Critical Tables," National Research Council, McGraw-Hill Book Company, Inc., New York, N. Y., vol. 2, fig. 12, 1927, p. 157.

7 "Carbon Hydrogen Ratios of Distillate Fuel Oils," by S. P. Cauley and E. B. Delgass, *Oil and Gas Journal*, vol. 46, 1947, pp. 83–84.

8 "Mineral Market Report," No. MMS 1666, Bureau of Mines, Dec. 6, 1948.

Discussion

F. T. Hague.[3] The total fuel cost for an industrial gas turbine must not exceed that of a Diesel engine if the gas turbine is to be a competitive engine. This means that it must burn residual oil. Residual oils as a group have two characteristics which must be controlled to insure satisfactory use. These are (1) the compatibility or freedom from sludge formations when batches of different origin and history are mixed together; and (2) the ash content of metallic salts which adheres to turbine blading and renders units inoperative.

In reference to the use of residual oil, it should be recognized that any gas-turbine manufacturer can develop a combustor for efficient combustion, and that the turbine-blade deposits, resulting from the use of an unsuitable batch of oil, are independent of the combustor design or the name of the turbine manufacturer. The problem of avoiding the use of unsuitable oil in a gas turbine has the characteristics of an industry problem, in that it is in the interest of the customer, the oil supplier, and the turbine manufacturer that the impossible not be attempted.

The industrial gas turbine gives promise of being a continuous outlet for a nominal percentage of residual oil. The oil industry's interest in developing this outlet for the otherwise surplus "what's left" of the refining process is commendable in all respects. Let's consider the wealth of information at their disposal to do something constructive for the gas-turbine fuel-oil problem.

The oil suppliers know the field origin and the processing history of their residual oils. They recognize the typical ash content of the crudes from specific oil fields, such as the objectionable vanadium pentoxide from South America, and the sodium and calcium sulphates from domestic fields. We understand they recognize the impossibility of mixing refinery stocks from straight-run catalytically or thermally cracked processes without incurring incompatibility. This general type of information and experience is the present basis for preparing Navy Special residual oil.

The gas-turbine group must now become sufficiently articulate to tell the oil industry what it requires of a gas-turbine residual oil. Once the oil industry is convinced there is a gas-turbine residual-oil problem and takes the necessary steps to segregate and market a suitable "gas-turbine residual oil," it will be using its potential technical knowledge and experience most commendably. There is every indication that the various oil suppliers should get together and pool their experiences so that the availability of a gas-turbine residual oil fuel is a reality of tomorrow. Diesel oil fuel is closely controlled by specification because the Diesel and oil industries recognize the common-sense economics of using a suitable fuel. The gas-turbine industry is yet in the embryo stage, but the same common sense should rapidly mature the thinking on its fuel problem. Customers aren't going to like it otherwise.

Ashton Howarth.[4,5] More emphasis, it is believed, should be placed on ash content and its effects, as it constitutes one of the principal objections militating against the use of residual-type fuels in gas turbines. Serious as the corrosion problem may be, it is probable that susceptibility to attack by and deposition of vanadium compounds and other salts also are of great importance. Part of this problem is not much different from that encountered in steam-boiler practice where deposition of materials in the forms of sulphates, glasses, and oxides in superheater banks restricts the passage of combustion gases through the boiler and interferes with its operating efficiency.

The authors cite an example where a high-ash-content fuel produced no serious effect. Type of constituent is even more important than the amount; a low ash content of certain constituents can be much more offensive than higher amounts of other materials. Published analyses show one of the most bothersome constituents to be sodium sulphate.

Hughes and Voysey[6] describe in great detail the ash and deposits found on gas-turbine blades and show that sodium is the most likely to form harmful deposits. They also discuss the various processes by which the ash adheres to the blades and possible means of dealing with this problem.

In connection with possible corrective measures, it is not to be overlooked that if the ash constituents are present in the vapor phase, their removal by mechanical means such as filtration is impractical.

E. W. Jerger.[7] It is indeed true that the C/H ratio is difficult to obtain by a method of burning the fuel and analyzing the combustion products. At present there is not available gas-analyzing equipment to enable one to make an extremely precise exhaust-gas analysis in oil burning. Then, in the calculation of the C/H ratio it would have to be assumed that the fuel is composed of only carbon and hydrogen. Ordinary heavy commercial distillates, even if low in sulphur content, rarely have greater than a 97 per cent combined carbon-hydrogen content. The aniline point and the (UOP) *K* factor, in fact, both are a measure of the paraffinicity of the oil. Hougen and Watson[8] have shown that the *K* factor, average boiling point, and API gravity are related to the molecular weights of petroleum fractions. In general, paraffinic fractions have higher molecular weights than aromatic fractions with corresponding lower *K* factors, at approximately the same average boiling point. A correlation is also given between the hydrogen content and the *K* factor plus one other property. It seems that the C/H ratio with the molecular weight would indicate the general chemical structure of the compound.

---

[3] Consulting Engineer, Steam Division, Westinghouse Electric Corporation, Lester Branch, Philadelphia, Pa.

[4] Superintendent, Petroleum Division, Naval Boiler and Turbine Laboratory, Philadelphia Naval Shipyard, Philadelphia, Pa.

[5] The opinions or assertions contained herein are the private ones of the writer and are not to be construed as official or reflecting the views of the Navy Department or the naval service at large.

[6] "The Considerations Dealing With the Formation of Deposits in Gas-Turbine Plant," by Hughes and Voysey, *Journal of the Institute of Fuel*, April, 1949, p. 197.

[7] Assistant Professor, Department of Mechanical Engineering, Iowa State College, Ames, Iowa, Jun. ASME.

The 50 per cent distillation temperature should only be used as the average boiling point for petroleum fractions of the same crude. A mixture of different-gravity distillates will exhibit a wide boiling range.

The writer takes exception to the correlation cited between Diesel index and cetane number. While the aniline point correlates well with the cetane number, the usual differences in API gravity for fuels taken from the same crude destroys a good correlation between the Diesel index and cetane number.

One property the author didn't mention that may have a bearing on the combustion-chamber design, from a combustion time standpoint, is the heat of vaporization. The heat of vaporization has also been empirically related to API gravity and boiling point, and can of course be calculated with the classical Clapeyron equation.

In regard to the mention of preheating fuel oil supplied to a burner; it has been the writer's experience that the preheat temperature should correspond to an optimum viscosity. The optimum viscosity seems to be a matter of droplet size, and is best found by experiment. Preheating a heavy residual No. 6 fuel oil to approximately 180 to 190 F seems to give the best results.

AUTHORS' CLOSURE

From the limited experimental and operational data available, the fuel problems associated with the use of residual fuels in gas turbines are by no means clear cut. Both railroads and merchant vessels, which are substantial consumers of residual fuels, have not suffered incompatibility difficulties though their fueling points cover a wide geographic area. The insistence of the Navy Department on a compatibility specification for its boiler fuels is probably based on the much greater area its vessels would be expected to cover as well as the necessity for minimizing operating difficulties however remote their possibility. Consequently, very few instances of incompatibility are anticipated in the utilization of residual fuel oils in gas turbines.

Hughes and Voysey in some commendable experiments did show that certain elements in fuel oils contribute more to turbine deposits than others. In their work with a 500-hp experimental gas turbine, deposit formation on the turbine blading was found to be progressive. Other turbine users, however, found that some deposits reached a maximum thickness and then remained constant indicating that operating conditions and turbine design were factors related to this phenomenon. The recent work of Buckland and Barkey on a 4800-hp commercial gas-turbine unit apparently did not lead into any deposit problems in the turbine even though residual fuels of varying ash contents were used. Corrosion, however, was encountered with a high-ash fuel containing vanadium. It is evident that more operating data on industrial gas turbines are needed to evaluate the effects of ash.

It is difficult to follow Jerger's contention that the 50 per cent distillation temperature should be used as the average boiling point only for petroleum fractions of the same crude. The C/H ratio correlations cited in the paper were developed from a series of 22 distillate fuels covering different crude oils and different refinery processing methods. The average deviation of estimated from actual C/H ratio was found to be 0.89 per cent and the maximum deviation was 3.27 per cent.

The aniline point of Diesel fuels has not been found to correlate well with the cetane number. An estimation procedure utilizing only the 50 per cent boiling point and the API gravity[9] of a fuel has been found to give acceptable estimates of cetane numbers for most fuels. The Diesel index correlation was presented because of its use as an alternate to the cetane-number requirement in a number of specifications.

---

[8] "Chemical Process Principles," Part 1, by Hougen and Watson, John Wiley and Sons, Inc. New York, N. Y., 1943.

[9] "The Correlation of Cetane Number With Other Physical Properties of Diesel Fuels." Report by the Diesel Index Panel of Standardization Subcommittee No. 4, Gas Diesel & Fuel Oils, *Journal of the Institute of Petroleum*, vol. 30, 1944, p. 193.

Combination of Heat Sources and Sinks by the Method of Superposition[1]

By F. H. BRIDGERS,[2] LOS ALAMOS, NEW MEXICO

A superposition method is proposed whereby the temperature pattern around a single buried heat source at a given temperature is used to arrive at the pattern obtained when several heat sources at the same temperature are placed at any distance apart. An equation is derived to give the "strength ratio," which is defined as the ratio of heat dissipated by one of a series of buried heat sources of equal temperature to the heat dissipated by a single heat source at the same temperature. This equation is plotted for a specific case to show the "strength ratio" as a function of heat-source spacing. Experimental temperature-distribution patterns around a heated tube buried in a box of sand are shown for the center tube of a group of three tubes. Although the proposed superposition method is subject to an inherent error, which may be severe at very close spacing of heat sources, the agreement between experimental and calculated results for the cases investigated in this work is good.

NOMENCLATURE

The following nomenclature is used in the paper:

A = distance from center of tube to "near" surface of equivalent slab

$$A = a + \frac{k_a}{f}$$

a = distance from center of tube to near surface in actual slab
B = distance from center of tube to "far" surface of equivalent slab; $B \gg A$
d = distance between tubes (center to center)
f = combined radiation-convection heat-transfer coefficient
$f(x, y)$ = mathematical function defined by Equation [1*d*]
$f(x, 2A\text{-}y)$ = mathematical function defined by Equation [1*e*]
k = thermal conductivity
k_a = thermal conductivity of equivalent slab
ln = natural logarithm
q = heat flux per unit length of source
r = distance from line source or sink
r_0 = radius of cylindrical heat source
r_1, r_2, r_3, r_4 = distances from a particular line source or sink
$t_{x,y}$ = temperature of a point in a slab
t_w = temperature of tube surface
t_e = effective temperature of environment on heating side of slab; approximately equal to $\frac{t_a + t_s}{2}$
t_a = air temperature on heating side of slab
t_s = average surface temperature of surroundings on heating side of slab
t_o = effective temperature of environment on nonheating side of slab
α = constant defined by Equation [1*a*]
β = constant defined by Equation [1*f*]
γ = constant defined by Equation [1*i*]
Δ = constant defined by Equation [1*b*]
θ = temperature potential (point temperature minus sink temperature)
$\theta_0, \theta_1, \theta_2$ = temperature potentials due to a particular line source or sink
θ_p = temperature potential due to a combination of line sources or sinks
η = "strength ratio" defined as ratio of heat dissipated by one of a series of buried heat sources of equal temperature to heat dissipated by a single heat source at same temperature
π = mathematical constant, = 3.14159 . . .
$\psi(x, y)$ = mathematical function defined by Equation [1*e*]
$\psi(0, \text{-}B)$ = constant defined by Equation [1*h*]
$\psi(r, 0)$ = constant defined by Equation [1*g*]

INTRODUCTION

The solution of heat-transfer problems involving the two-dimensional steady flow of heat from a single heat source in a solid medium has been proposed by Carslaw (1)[3] and others (2, 3). The various solutions are exact only when the boundaries of the solid mediums are isothermal sinks. For the case where the boundary of the solid medium is not isothermal, such as a pipe or cable buried in a wall, the solution is greatly complicated because of the complexity of establishing exact boundary conditions. Schofield (4) has shown that approximate solutions of such problems are possible by establishing simplified approximate boundary conditions. If there exists, instead of a single heat source, a series of heat sources at the same temperature buried in a slab, the mathematics of the problem becomes much more involved. This work will deal principally with the approximate solution of the type of problems where a series of cylindrical heat sources at the same temperature are buried in a solid medium. A practical example of such a case would be that of a floor-heating panel having hot water pipes buried in concrete. The boundary of the solid (panel surface) for the case considered will not be an isothermal sink, but will have a surface temperature distribution being governed by the balance at each point of the heat conducted to the surface from heat sources and that transferred from the surface by convection and radiation.

Vanderweil (5) has derived an equation to give the two-dimensional temperature distribution in a slab containing equally spaced heated tubes at the same temperature. His equation holds only when the tubes are located very near to one of the bounding surfaces of the slab, so that the percentage of heat flow

---

[1] Based on a Master's Thesis, 1948, School of Mechanical Engineering, Purdue University.

[2] Engineer, Charles S. Leopold, Consulting Engineering Firm, Philadelphia, Pa.

Contributed by the Heat Transfer Division and presented at the Fall Meeting, Erie, Pa., September 28–30, 1949, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 49—F-13.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

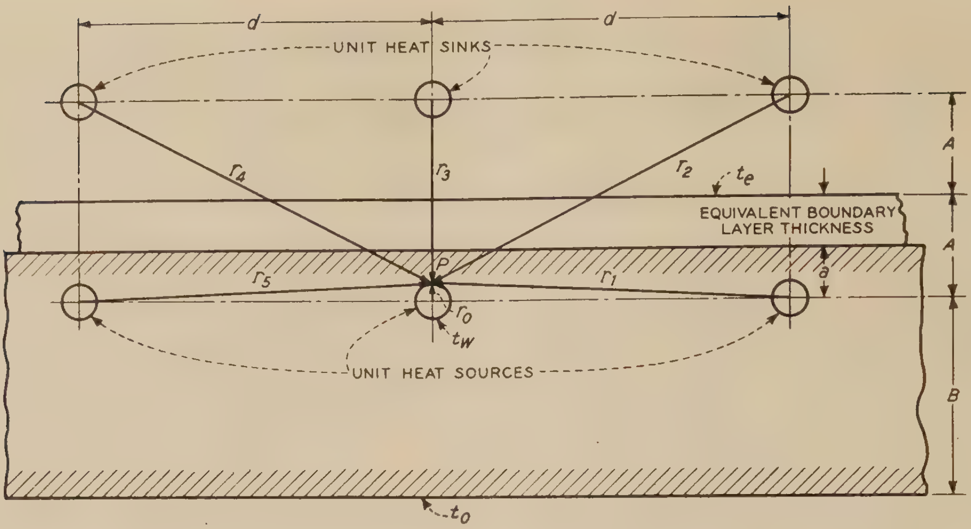


Fig. 1 Boundary Conditions for Heat Sources in a Slab

through the opposite surface is small compared to that through the near surface.

He established simplified boundary conditions by replacing the actual slab, including the convection-radiation heat-transfer film, with a slab of equivalent thickness of a common conductivity, Fig. 1. His final equation is a complex function of nine variables expressed in the form

$$t_{x,y} = \alpha\psi(x, y) + \beta y + \gamma \ldots\ldots\ldots\ldots\ldots [1]$$

where

$$\alpha = \frac{1}{\Delta}\,[A(t_w - t_0) + B(t_w - t_0)] \ldots\ldots\ldots [1a]$$

$$\Delta = (A + B)\psi(r_0, 0) - A\psi(0, -B) \ldots\ldots\ldots [1b]$$

$$\psi(x, y) = f(x,y) - f(x, 2A - y) \ldots\ldots\ldots [1c]$$

$$f(x, y) = \frac{1}{2}\ln\left[\sin^2\frac{\pi x}{d}\cosh^2\frac{\pi y}{d} + \cos^2\frac{\pi x}{d}\sinh^2\frac{\pi y}{d}\right] \ldots [1d]$$

$$f(x, 2A - y) = \frac{1}{2}\ln\left[\sin^2\frac{\pi x}{d}\cosh^2\frac{\pi(2A - y)}{d} + \cos^2\frac{\pi x}{d}\sinh^2\pi\frac{(2A - y)}{d}\right] \ldots\ldots [1e]$$

$$\beta = \frac{1}{\Delta}\,[\psi(r_0,0)(t_e - t_0) + \psi(0, -B)(t_w - t_e)] \ldots\ldots [1f]$$

$$\psi(r_0, 0) = \ln\sin\frac{\pi r_0}{d} - \frac{1}{2}\ln\left[\sin^2\frac{\pi r_0}{d}\cosh^2\frac{\pi 2A}{d} + \cos^2\frac{\pi r_0}{d}\sinh^2\frac{2\pi A}{d}\right] \ldots\ldots [1g]$$

$$\psi(0, -B) = \ln\sin\frac{\pi B}{d} - \ln\sinh\frac{\pi}{d}(2A + B) \ldots\ldots [1h]$$

$$\gamma = \frac{1}{\Delta}\,[A\psi(r_0, 0)t_0 + B\psi(r_0, 0)t_e - A\psi(0, -B)t_w \ldots [1i]$$

The origin of the x, y-co-ordinates is taken as the intersection of the horizontal and the vertical center lines of a tube. This equation is applicable only when the distance B, from the center of the tubes to the far surface is equal to or greater than the distance d between tubes, Fig. 1.

It is obvious that the foregoing equation would be tedious to apply to a given problem. The method proposed in this paper provides a simpler solution to this same type of problem.

The purpose of this work is to investigate the possibility of adapting the principles of superposition for solutions of Laplace's equation to the case of heat flow from a tube embedded in a slab. These principles of superposition are well known in various fields of scientific work, such as hydrodynamics, electrostatics, and transmission-line theory. In the field of heat transfer by conduction the steady-state temperature potential θ, satisfies the Laplace equation for two-dimensional heat flow in a region where no heat is generated or stored, and can be expressed as

$$\frac{\partial^2\theta}{\partial x^2} + \frac{\partial^2\theta}{\partial y^2} = 0 \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

The analytic function of a complex variable and its real and imaginary parts are functions, respectively, of equipotentials and constant-flow lines (6). Potential fields can be combined by simply adding the potentials of the individual fields (7). Such a procedure for combining thermal potential fields (isothermals) is the subject of this investigation. By using the isothermals from a single buried tube, one can arrive at the combined pattern of isothermals that would be obtained for the case of a series of tubes at the same temperature.

Adaptation of Superposition Method for Combining Heat Sources

The temperature-potential function of a line source in an infinite slab, bounded on one side by an isothermal plane, can be determined by the method of conformal transformation so that the Laplace equation is satisfied (4). This function has been found to be

$$\theta = \frac{q}{2\pi k}\ln\frac{r_2}{r_0} \ldots\ldots\ldots\ldots\ldots\ldots [3]$$[4]

Applying the procedure of combining potential fields by simply adding the potentials of the individual fields, the temperature potential at a point P in a slab with two sources, Fig. 2, would be

---

[4] This equation has been more directly derived by Boelter and coworkers (6).

$$\left.\begin{aligned}\theta_p &= \theta_0 + \theta_1 = \frac{q}{2\pi k}\ln\frac{r_2}{r_0} + \frac{q}{2\pi k}\ln\frac{r_4}{r_3}\\ \theta_p &= \frac{q}{2\pi k}\ln\frac{r_2 r_4}{r_0 r_3}\end{aligned}\right\}\quad\ldots\ldots[4]$$

If the isothermal at r_0 represents the surface of a pipe with constant water temperature, the temperature potential at P will remain constant at θ_0 regardless of how close line source B approaches source A; therefore Equation [4] does not hold. It is forseeable that the strength, q (B hr^{-1} ft^{-1} of length), of each individual source, when combined with other sources in the solid at the same temperature will be less than the strength of the individual source alone in the solid. In other words, if two pipes at the same temperature were placed close together in a solid, each individual pipe would dissipate less heat than if one of the pipes were alone at the same temperature. Therefore, by fixing the temperature potential at r_0, for the case of two sources close to one another, Equation [4] should be written in the form

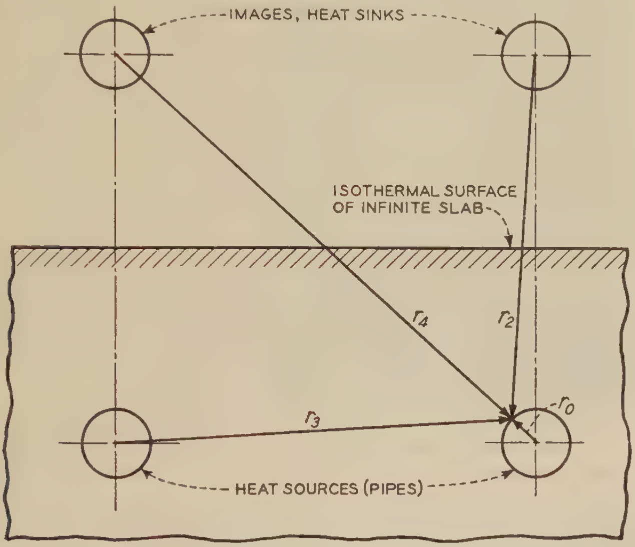


Fig. 2 Combination of Two Heat Sources in an Infinite Slab

$$\theta_p = \theta_0 = \eta\,\frac{q}{2\pi k}\ln\frac{r_2 r_4}{r_0 r_3}\ldots\ldots\ldots\ldots\ldots[5]$$

where η is the "strength ratio," defined as the ratio of heat dissipated by one of the two tubes when together, to that dissipated by the same tube when by itself and at the same temperature.

Solving for η in Equation [5]

$$\eta = \frac{\theta_0 2\pi k}{q\ln\dfrac{r_2 r_4}{r_0 r_3}}\ldots\ldots\ldots\ldots\ldots\ldots[6]$$

Solving for q in Equation [4] and substituting in Equation [6], we obtain

$$\eta = \frac{\theta_0}{\theta_0 + \theta_1}\ldots\ldots\ldots\ldots\ldots\ldots[7]$$

In Equation [7], θ_0 is fixed as the difference between the temperature of the source and the temperature of the sink, while θ_1 is evaluated by means of Equation [3]. Equation [3] is given in terms of x, y-co-ordinates by Boelter and coworkers (6). Equation [7] provides a good approximation for the change in strength q (B hr^{-1} ft^{-1} of length) of two pipes as a function of their distance apart in a slab. For the limiting case of two pipes when the distance apart approaches zero, η will approach 0.5, as would be expected for equal temperature pipes. As the distance apart increases, η will approach 1.0. The foregoing treatment is not limited to two pipes but could be applied to a series of pipes, and the decrease in strength of a single pipe could be similarly computed.

In applying the analytical treatment to an air-cooled slab with a series of embedded tubes, only an approximate solution can be obtained because of the complexity of establishing true boundary conditions. Equation [3], which was used to obtain Equation [7], holds only when the slab is bounded by an isothermal plane. Such is not the case for a finite slab with embedded tubes, but rather there is a definite surface-temperature gradient between the adjacent tubes. Vanderweil proposed approximate boundary condition in developing his theoretical equation which we may employ in the analytical approach of this treatment. He considered the actual slab, including the combined surface radiation-convection film, to be replaced by an equivalent slab of constant conductivity, which has the same resistance to heat flow as the actual slab. Such an equivalent slab is shown in Fig. 1 where the heat flow through the back side of the slab is neglected. If we assume the strength of three sources to remain unchanged upon combining, the temperature at point P, Fig. 1, would be

$$\theta_p = \frac{q}{2\pi k}\left[\ln\frac{r_3}{r_0} + \ln\frac{r_2}{r_1} + \ln\frac{r_4}{r_5}\right]\ldots\ldots\ldots[8]$$

The strength ratio for the center tube can be obtained in the same manner as used to obtain Equation [7]

$$\eta_A = \frac{\theta_0}{\theta_p}\ldots\ldots\ldots\ldots\ldots\ldots[9]$$

For an infinite number of equally spaced tubes the temperature potential at any point in the slab will be

$$\theta = \eta\,\frac{q}{2\pi k}\left[\ln\frac{r_1}{r_2} + \ln\frac{r_3}{r_4} + \ln\frac{r_5}{r_6} + \ldots\right]\ldots[10]$$

where r_2, r_4, r_6 are the distances from the sources to the point, and r_1, r_3, and r_5 are the distances from the imaginary sinks to the point.

Fig. 3 shows a plot of the strength ratio versus tube spacing for the center pipe of a three-pipe combination at a 2-in. depth of burial in sand. The values for this plot were obtained by considering only three pipes and using Equations [8] and [9]. It is noteworthy that this curve will hold theoretically for any source temperature when employing Equations [8] and [9], provided bound-

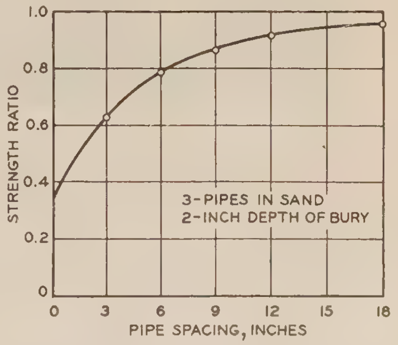


Fig. 3 Strength Ratio as Function of Pipe Spacing for Center Pipe of Three-Pipe Combination

ary conditions are assumed to remain constant. Whether or not the curve will be valid for a wide temperature range in actual practice is a matter of experimental confirmation. In an actual slab where there is a series of tubes, it is obvious that when the pipes are spaced closely, there will be more than two pipes which will affect the temperature potential θ_p, and therefore the strength ratio. In any arrangement of a series of sources, one must take into account the effect of all those sources which influence the potential θ_p, Fig. 1.

It should also be mentioned that the smaller the depth of tube burial and the closer the tube spacing, the greater will be the error introduced by using this method based on line sources and sinks. In using the treatment based on line sources, the assumption was made that the isothermal at a distance r_0 (through point P, Fig. 1), from the source is circular and therefore represents the pipe surface. The isothermal through point P will not remain circular as two sources come close together, or as the distance to the surface becomes small, but will become distorted. The greater the distortion, the greater will be the error introduced.

An illustrated solution of the foregoing treatment is given in Fig. 4. This figure shows a method of using the temperature gradient along the horizontal center line from a single buried tube to obtain the gradient that would occur along the horizontal center line for a three-tube combination. For example, the resulting temperature potential at point P is given by

$$\theta_p = \eta_A \, \theta_{A_p} + \eta_B \, \theta_{B_p} + \eta_C \, \theta_{C_p} \qquad [11]$$

where

$$\eta_A = \frac{\theta_0}{\theta_A}$$

and

$$\eta_B = \eta_c = \frac{\theta_0}{\theta_B} = \frac{\theta_0}{\theta_C}$$

Temperature potentials θ_0, θ_p, θ_A, θ_B, θ_C, θ_{A_p}, θ_{B_p}, and θ_{C_p} are shown in Fig. 4. The resulting gradient along the horizontal center line connecting the three tubes is obtained by adding the potentials at a number of points as just shown. In this way, the temperature-distribution pattern of a single tube is used to obtain the temperature pattern of a combination of tubes for any particular tube spacing. If the experimental temperature gradients from the single tube are used in applying the illustrated superposition method, the results should be more accurate than the results obtained from an analytical solution where boundary conditions are approximated.

Experimental Equipment and Procedure

The experimental phase of this work consisted in obtaining data on the temperature-distribution patterns around single and multiple buried tubes, in order that the proposed superposition method could be checked. Temperature patterns were obtained for the following conditions:

1 Single tube as the only heat source in the solid.

2 The tube combined with either one or two other heat sources (tubes at the same temperature at a distance of 6 in. from centers).

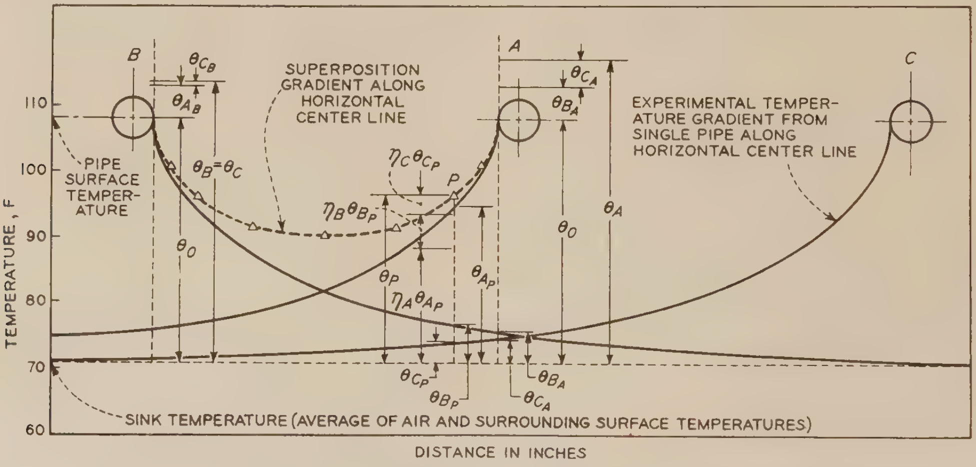


Fig. 4 Illustrated Application of Superposition Method for Case of Three Heat Sources

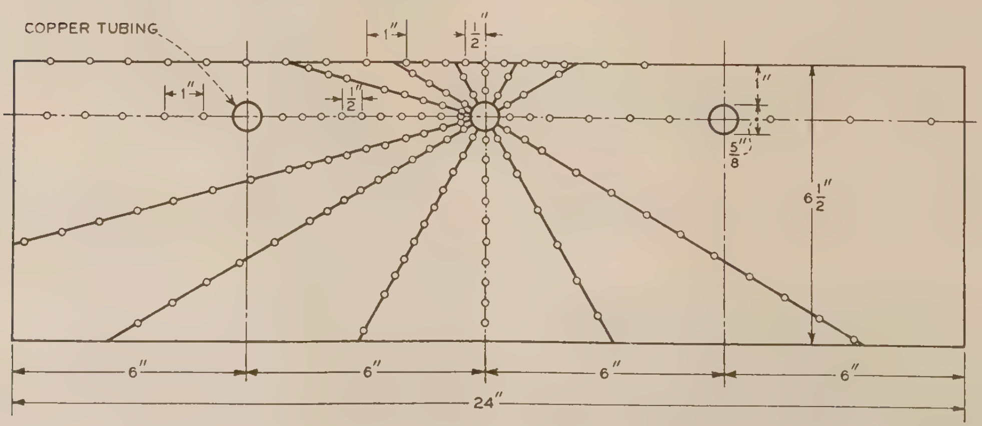


Fig. 5 Thermocouple Arrangement at Center Cross Section of Slab

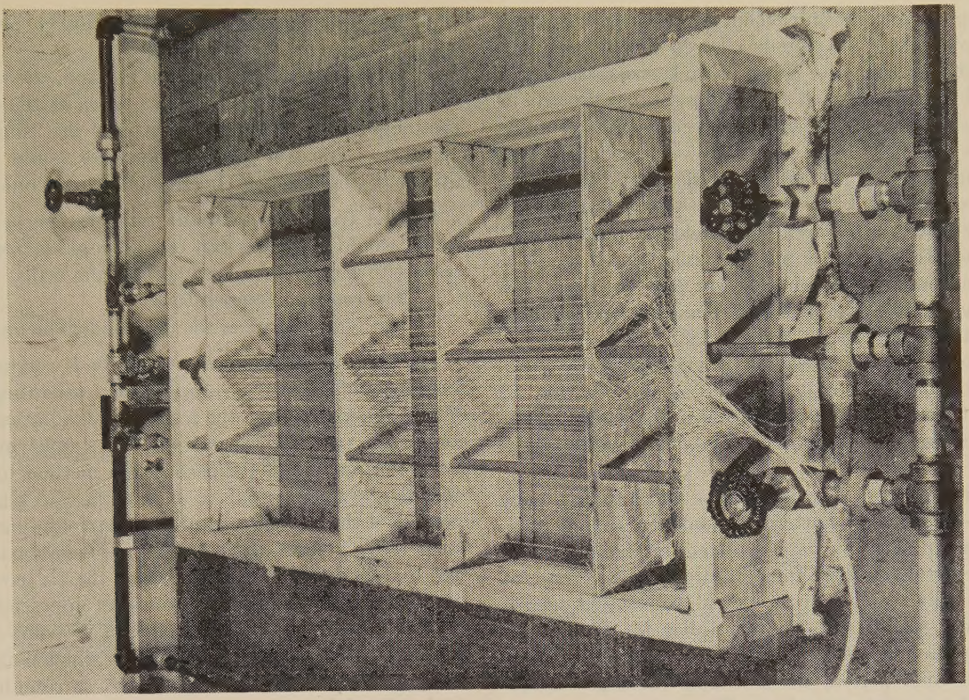


Fig. 6 Experimental Equipment Showing Arrangement of Thermocouples

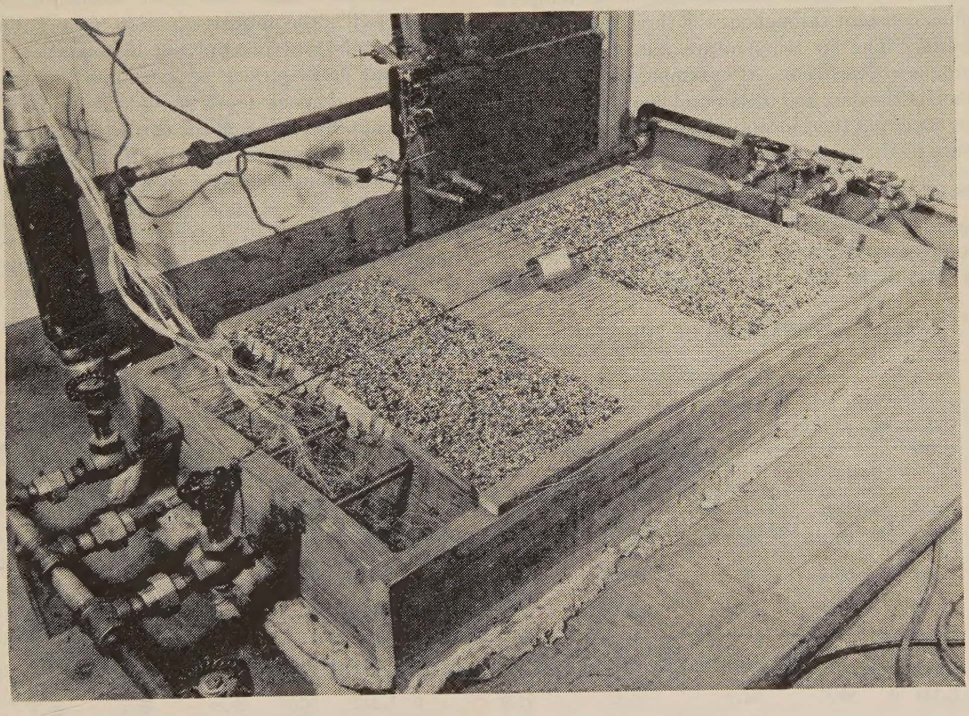


Fig. 7 Experimental Equipment Showing Slab Under Test

3 In both of these cases, three depths of burial were used, namely, $^1/_2$ in., 1 in., and 2 in.

The experimental equipment used to obtain the temperature pattern around the buried tubes consisted of a box containing three $^5/_8$-in-OD copper tubes buried in sand at various depths. Copper-constantan thermocouples were installed in a radial pattern around the middle tube at the center cross section of the slab. Fig. 5 shows the typical arrangement of the thermocouple for the 1-in. depth of tube burial. The thermocouples indicating the tube temperature were soldered on top of each tube. All thermocouple leads were brought out parallel to the tubes. Fig. 6 shows the box containing the tubes and thermocouples before the sand was poured. The thermocouple leads were held in place by four $^1/_4$-in. plywood boards which had small holes drilled to accommodate the leads, Fig. 6. Dry sand which had been sifted through a 16-mesh screen was poured into the center section, care being taken to prevent movement of the thermocouples. The sand was not packed, but merely allowed to run freely into the box. Expanded vermiculite (loose particle size) was poured into the two end sections on each side of the sand to provide insulation to reduce the heat flow in the direction parallel to the pipes. Thus the heat flow in the sand slab was essentially two-dimensional. Fig. 7 shows the assembled slab with the copper tubes buried at a depth of 1 in. Additional thermocouples outside the box were placed as follows: One shielded thermocouple 2 in. above the sand over the center pipe to record the air temperature; two thermocouples between the box and the floor, one of which was above and the other below the blanket-type insulation which was placed between the box and the floor.

The thermocouple emf was measured by a semiprecision portable potentiometer which could be read accurately to within 0.2 deg. During all tests the water temperature in the tubes and the ambient room air temperature were maintained within approximately 1 deg.

Three tests were run for each depth of tube burial to determine the temperature-distribution pattern around the center tube. The first test was run with the water directed through the center tube only. Immediately after the temperatures were recorded for the first test, the valve on one of the outside tubes was opened and the water was passed through two tubes for the second test. Water was passed through all three tubes for the third test. During the equilibrium period, thermocouple readings were recorded along the horizontal center line. When the temperatures at points along this line ceased to change with time, it was assumed that steady-state conditions existed. The water temperature was maintained constant throughout the three tests for each depth of tube burial. The air temperature was also constant throughout the progress of the tests. After temperatures throughout the cross section of the box, and the air and floor temperatures were recorded, an exploring thermocouple was used to obtain the surface temperature of the surrounding walls and ceiling.

DISCUSSION OF RESULTS AND CONCLUSIONS

The results are summarized in Table 1, and Figs. 8, 9, and 10, which show a comparison between experimental temperatures, the temperatures obtained by using the superposition method and temperatures computed by the Equation [1] proposed by Vanderweil.

In applying the superposition method, the temperature potential at any point in the slab was taken as the difference between the temperature at a given point and the "sink" (ambient) temperature. The true sink temperature would be difficult to obtain as it would depend on the following temperatures:

1 The true mean radiant temperature of the slab surface, which would be a function of the shape factor of the slab with respect to each of the surrounding room walls.

2 The room air temperature.

3 The temperature of the floor under the test box.

The true sink temperature would be a weighted average of the foregoing temperatures, proportioned according to the per cent of heat dissipated from the test box by radiation, convection, and conduction (through the bottom). A simplifying assumption was made in applying the superposition method by considering the sink temperature equal to either the air temperature or the average of the air temperature and surrounding surface temperatures, which in most cases would differ from the true sink temperature by less than 2 or 3 deg F, using the air temperature alone as the sink temperature might cause an appreciable error in some cases. Table 1 shows a comparison of point temperatures obtained by the superposition method in which both the air temperature and the average of the air and surrounding surfaces were used as the sink temperature.

The superposition-point temperatures are also compared with experimental values and point temperatures calculated using Vanderweil's theoretical equation (see Table 1). Vanderweil recommends that in applying his equation for calculating the minimum panel-surface temperature (midway between tubes) that the equivalent depth of the tube burial be multiplied by an experimentally determined correction factor; this correction factor would compensate for the increase in the equivalent boundary layer at that point due to a smaller temperature difference between the air and panel surface. In computing the temperatures shown in Table 1, this correction factor was not used, which might account partly for the deviation of surface temperatures calculated by this method compared with the experimental data.

TABLE 1 COMPARISON OF POINT TEMPERATURES OBTAINED BY EXPERIMENTAL, SUPERPOSITION, AND VANDERWEIL METHODS

One-Half Inch Depth of Tube Burial - Three Pipes - 107.2 F. Surface Temperature

ALONG HORIZONTAL CENTER LINE						AT SURFACE OF SLAB					
			OBTAINED BY SUPERPOSITION METHOD						OBTAINED BY SUPERPOSITION METHOD		
Point X	Y	Experimental Values F	Based on sink temp. t_a, F	Based on sink temp. t_e, F	Obtained from Vanderweil Equation F	Point X	Y	Experimental Values F	Based on sink temp t_a, F	Based on sink temp t_e, F	Obtained from Vanderweil Equation F
1.312	0	90.0	90.1	90.1	90.8	0	0.812	92.2	93.1	93.1	94.9
2.312	0	83.8	84.6	84.6	86.2	1.5	0.812	82.8	83.6	83.6	87.4
3.0	0	83.0	83.6	83.6	85.2	3.0	0.812	79.0	80.5	80.5	84.3
One Inch Depth of Tube Burial - Three Pipes - 111.8°F. Surface Temperature											
1.312	0	96.0	94.3	95.0	95.9	0	1.312	90.0	88.8	89.6	89.5
2.312	0	91.0	88.9	89.7	91.6	1.5	1.312	85.3	85.6	85.3	86.4
3.0	0	90.0	87.8	88.3	90.8	3.0	1.312	82.0	81.7	81.7	84.0
Two Inch Depth of Tube Burial - Three Pipes - 108 F. Surface Temperature											
0.812	0	100.0	99.0	100.4	99.1	0	2.312	80.8	80.5	82.8	81.2
1.312	0	96.2	94.8	96.9	94.9	1.0	2.312	80.2	80.2	82.3	80.7
2.312	0	92.4	91.0	93.1	91.3	2.0	2.312	79.8	79.3	81.7	79.8
3.0	0	91.3	90.3	92.4	90.6	3.0	2.312	79.0	78.8	81.1	79.4

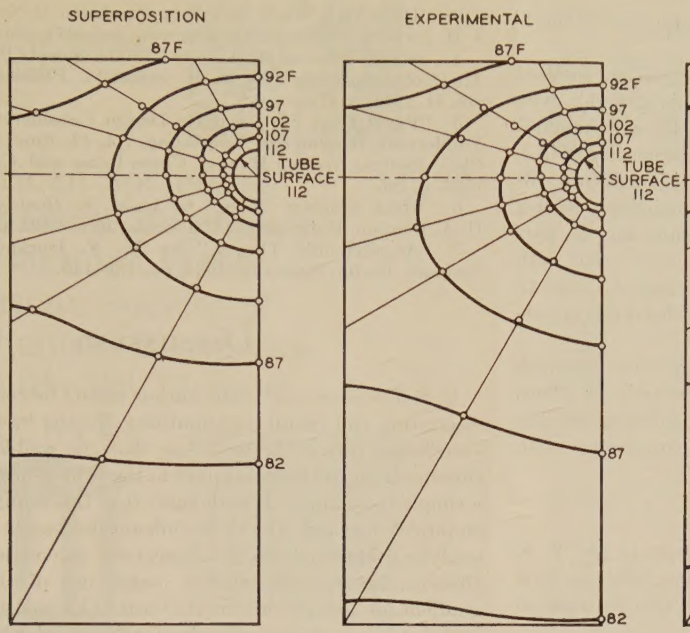


Fig. 9 Comparison of Isothermal Patterns Obtained From Experimental, Superposition, and Vanderweil Methods

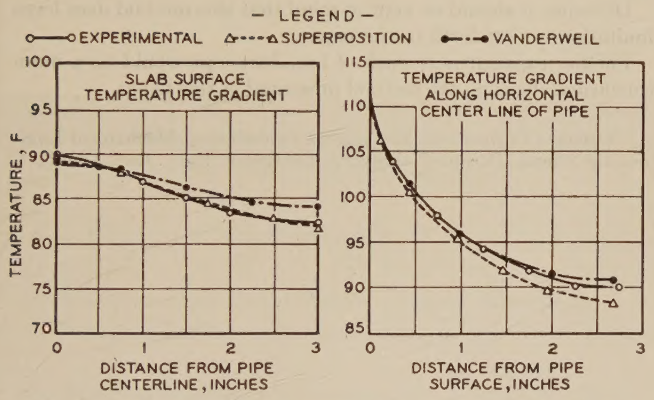


Fig. 8 Comparison of Experimental Gradients With Gradients Obtained by Superposition Method and Vanderweil Equation

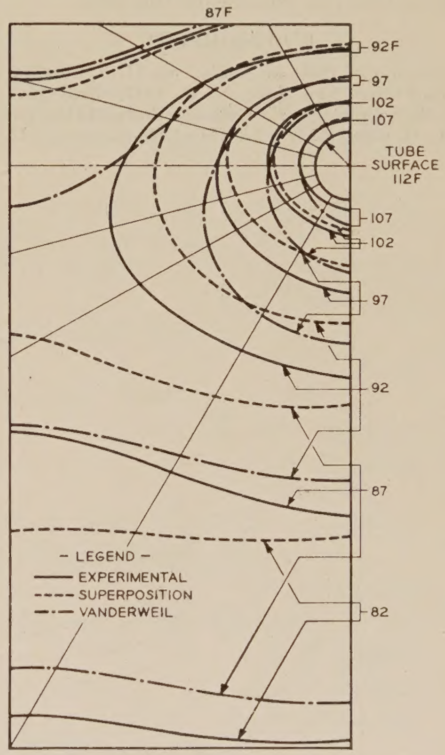


Fig. 10 Superimposed Comparison of Isothermal Patterns Obtained From Experimental, Superposition, and Vanderweil Methods

Fig. 8 shows a comparison of the experimental temperature gradients along the surface and horizontal center line between tubes, compared with those obtained by the superposition and the Vanderweil method. The average of the air and surrounding surface temperatures was taken as the sink temperature.

Fig. 9 shows a comparison of the temperature-distribution patterns around the center tube of a three-tube arrangement obtained by the experimental, superposition, and Vanderweil methods. The average of the ambient air and surrounding surface temperatures was taken as the sink temperature for the superposition method.

Fig. 10 shows the same isothermal patterns that are shown in Fig. 9, superimposed on each other so as to give a more direct comparison. It is seen that the agreement is very good between the three methods above the horizontal center line of the tube. Below the horizontal center line, the experimental isothermals fall about half way between those determined by the superposition and Vanderweil methods. The deviation of the superposition method from the experimental might be partly explained by experimental difficulties. The floor temperature under the test box could not be controlled and varied as much as 4 or 5 deg F. Therefore the true sink temperature for the experimental case, and the sink temperature used in applying the method were different. This caused a change in temperature potentials for points below the tube, and thus resulted in a deviation between the experimental and superposition isothermal patterns.

Results presented here are for the 1-in. depth of tube burial only. For the $^1/_2$-in. and 2-in. depths of burial the results of the superposition and Vanderweil methods deviated from the experimental results by about the same amount as for the case of the 1-in. depth of burial.

The results obtained here by using the superposition method cannot be taken as conclusive validation of this method, even though there is reasonably good agreement with the experimental data. It should be remembered that only one tube spacing (6 in.), one type of material (sand), and one value of temperature difference between water in the tube and air (approximately 37 deg F) were investigated. On the basis of the results for the particular case considered, it can be concluded, however, that both the superposition and the Vanderweil methods may be used to give a good approximation of the temperature-distribution pattern around one of a series of embedded tubes.

Equation [7], which has been derived to give the strength ratio, may have application in many heat-transfer problems such as the spacing of tubes in a heating or cooling panel, the arrangement of tubes for ground coils used in conjunction with the "heat-pump" and many similar applications.

Acknowledgments

The author wishes to express his sincere thanks to Dr. Y. S. Touloukian, Prof. F. W. Hutchinson, and Dr. Max Jakob for their encouragement and helpful suggestions throughout the course of this work.

The author is indebted to the Copper and Brass Research Association which made the funds available through the Purdue Research Foundation for carrying out this work.

BIBLIOGRAPHY

1 "Conduction of Heat in Solids," by H. S. Carslaw and J. C. Jaeger, Oxford Press, New York, N. Y., 1947, chapt. 15.

2 "The Heat Loss From a Cylinder Embedded in an Insulating Wall," by F. H. Schofield, *Philosophical Magazine*, vol. 12, 1931, pp. 329–349.

3 "Heat Flow When Boundary Condition Is Newton's Law," by J. H. Awbery, *Philosophical Magazine*, series 71, vol. 7, 1929, p. 1143.

4 "Steady Flow of Heat From Certain Objects Buried Under Flat Air-Cooled Surfaces," by F. H. Schofield, *Philosophical Magazine*, vol. 31, 1941, p. 471.

5 "Panel Heat Flow and Its Design Considerations," by R. G. Vanderweil, *Heating and Ventilating*, vol. 43, June, 1946, p. 63; also *Chase Radiant Heating Manual*, Chase Brass and Copper Company, 1945, p. 195.

6 "Heat Transfer Notes," by L. M. K. Boelter, H. V. Cherry, H. A. Johnson, University of California, July, 1942, pp. 4–10 and 8–8.

7 "Aerodynamic Theory," by W. F. Durand, editor, Julius Springer, Berlin, Germany, 1934, pp. 139–140.

Discussion

Y. S. Touloukian.[5] The author should be commended for his interesting and useful contribution. To the best of the writer's knowledge, this is the first time that the well-known theory of superposition has been adapted to the field of heat transfer, using a simple technique. It is thought that this semianalytical, semi-empirical method would be inherently better than a purely analytical approach on the equivalent convection radiation coefficient, because the author makes use of the experimental gradient for a single tube as the basis of his solution to obtain the surface temperature distribution for a series of tubes. This experimental gradient does take into account the exact influence of $(h_c + h_r)$.

Of course it should be kept in mind that this method does have limitations as set forth in the paper.

Further experimental work of broader scope would be a more conclusive check on the method presented in this paper.

---

[5] Assistant Professor of Mechanical Engineering, Mechanical Engineering School, Purdue University, Lafayette, Ind. Jun. ASME.